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May, 1941

Heating · Piping and Air Conditioning

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Journal Section
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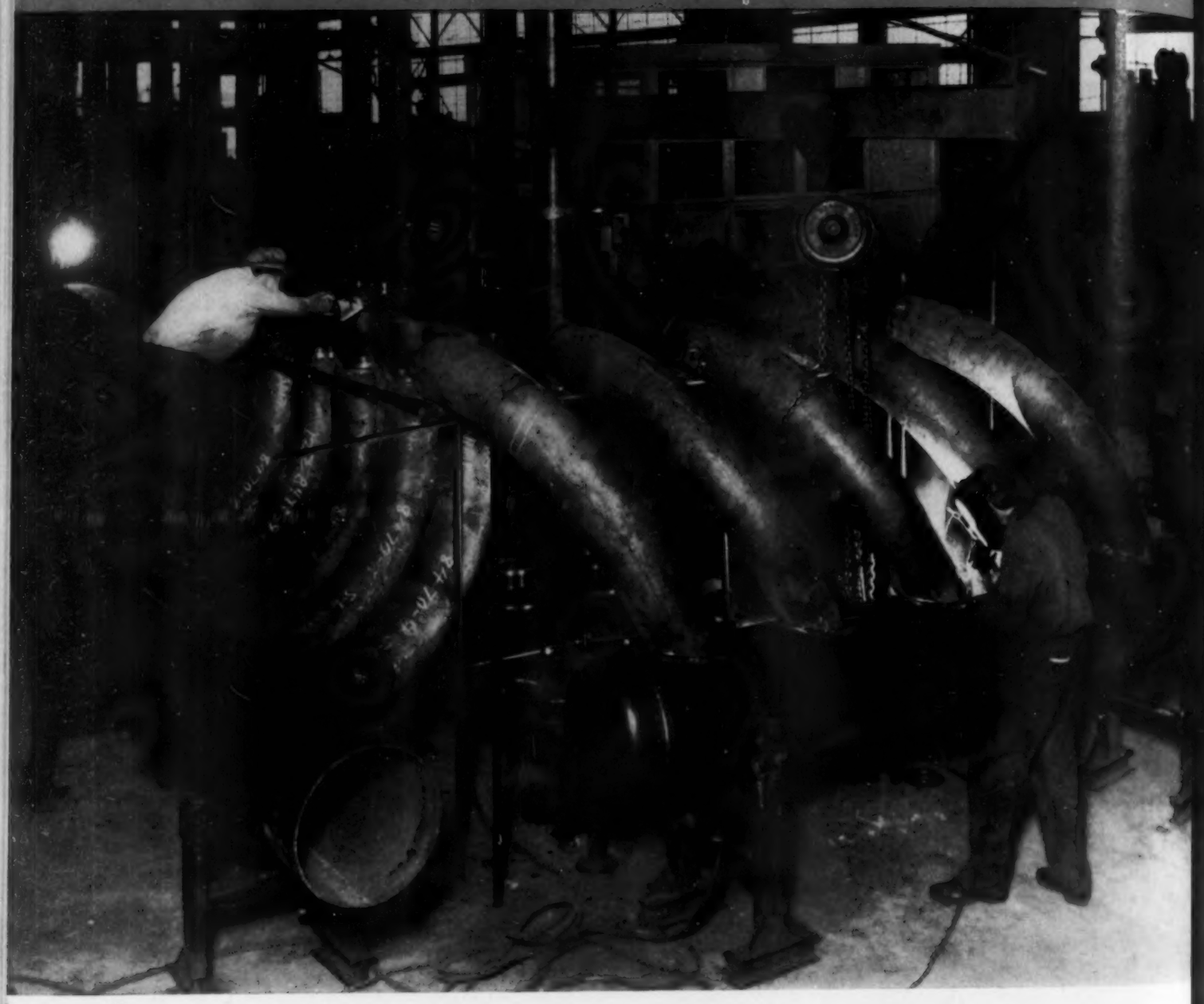
Here's What We Mean By "KNOWING HOW"!



THE plans called for special, heavy duty headers. The specifications were tough . . . predetermined strength . . . difficult curves and angles . . . exacting thermal characteristics . . . dimensional precision . . . delivery on a tight schedule.

Grinnell engineers have plenty of experience in handling these complicated assignments. Rough layouts of plant requirements were immediately interpreted in terms of alloy steel piping, accurate forming and welding, margins of safety, and underwriters' inspection. Conveniently located Grinnell plants then speedily prefabricated these and other subassemblies. And, promptly on delivery date, the complete installation was on the job . . . ready for erection.

This is only one of many examples of "knowing how" that prompts leading engineers to say, "Give the plans to Grinnell"! Write for detailed manual on Grinnell Prefabricated Piping. Grinnell Co., Inc., Executive Offices, Providence, Rhode Island. Branch offices in principal cities of United States and Canada.

PREFABRICATED PIPING BY **GRINNELL**

WHENEVER **PIPING** IS INVOLVED

The Editor's Pages

AIR CONDITIONING CAN SOLVE EXPECTED CRISIS IN IRON AND STEEL PRODUCTION

A prediction that air conditioning could go a long way toward solving the expected crisis in iron and steel production this spring was made by Dr. Willis H. Carrier, chairman of the board of Carrier Corp., in a radio interview last month. The American steel industry is equipped at present to produce roughly about 83,000,000 tons of steel annually, he said, and it is his understanding the industry would fall about 10,000,000 tons short of the total amount required for defense and consumer activity this spring. If each blast furnace could step up production by about 15 per cent, this would aid in solving the problem.

Dr. Carrier suggested the answer lies in air conditioning blast furnaces —to remove moisture from the air supplied them—instead of constructing new ones; particularly as steel producers might be unwilling to build new furnaces because, when the defense emergency is over, there might be no need for the increased output. At one steel plant in the South, in extreme summer weather, two separate economies have been realized by air conditioning. Out of the same furnace and plant—that is, out of the same capital investment and with the same labor—about 15 per cent more iron is produced than would be produced without dry blast. This means production of 115 tons as contrasted with the previous 100 tons. For the production of these 115 tons, about 90 per cent of the coke per ton, as compared with previous practice, is required.

According to the magazine *Time*, the cost of constructing new blast furnaces for a 10 per cent increase in output would be $115,000,000 and would require at least a year, while the cost of air conditioning existing blast furnaces to give the same increase in output would be but $16,000,000 and take two or three months. Thus, air conditioning would appear to be the logical answer.

"In addition to turning out a better quality pig iron, air conditioning is helping to reduce the cost of production and has resulted in an increase of approximately 15 per cent in the pig iron output," Dr. Carrier said. "The savings result from substantially lower consumption of coke. Coke requirements of the blast furnaces have been reduced by approximately 200 pounds per ton of pig iron, it is estimated. The need of less coke in the furnace also permits an increase in the amount of iron ore charged at one time. In the making of pig iron, furnaces require more than 4000 tons of air per day. The air conditioning system removes approximately 30 tons of water from this air each day."

[*Editor's note:* The first step in the conversion of iron into steel takes place in the blast furnace where the iron ore is freed from most of its associated impurities. Iron ore is a mineral combination of oxides of iron together with some silica, phosphorus, manganese, alumina and some other elements, with enough iron oxide present to make the mineral commercially economical to smelt.

The process consists of charging coke for fuel, limestone for flux, and iron ore into the top of the furnace and blowing in at the bottom a blast of heated air. The air burns part of the coke, the heat generated promotes chemical reactions and melts the charge. The molten iron is tapped from an opening at the bottom of the furnace into ladles which transport it to either the bessemer converters or open hearth furnaces for conversion into steel. In modern steel operations iron is not generally cast into "pigs" unless it is intended to be sold as merchant pig iron or is to be transported considerable distance.]

CONTRACTORS TO DISCUSS DEFENSE CONSTRUCTION

Among the subjects programmed for discussion at the 52nd annual convention of the Heating, Piping and Air Conditioning Contractors National Association is defense construction. Another topic to which the association members will give attention is that of sales problems, and reports of research in the industry on standardization, on trade promotion and on welding, on certification and on apprentice training will be made.

The meeting is to be held in the Hotel St. Francis in San Francisco, June 16-19. The first session will be a joint meeting with members of the American Society of Heating and Ventilating Engineers, who will be convening at the Palace Hotel in San Francisco the same week, at which time the Pacific Heating and Air Conditioning Exposition will also be held.

HEATING ENGINEER ON THE JOB

The National Safety Council's president's medal was awarded last month to Vernon Hupp, Chicago park district heating engineer, at a meeting of the park district board of commissioners. At the same time a National Safety Council sustaining certificate was given to park district patrolman Charles Koester. Both awards were for the successful artificial resuscitation of Hjalmer Swanson, whom Mr. Hupp saw floating face downward in Lake Michigan as he was driving along the outer drive last November.

Hupp lassoed Swanson with a rope and dragged him out of the water while a call was put in for a park district squad car. Hupp applied artificial respiration while waiting for the squad. He then turned him over to Patrolman Koester, answering the call, who worked on him for eleven minutes before normal breathing was restored.

ANNOUNCES INFORMATION ON CIVIL DEFENSE PROTECTIVE CONSTRUCTION

Publication of Civil Defense Protective Construction, a pamphlet prepared by the war department with the assistance and advice of other federal agencies, has been announced by the war department. This is the first of several civil defense pamphlets being prepared and it will be distributed by the division of state and local coöperation of the council of national defense.

The pamphlet presents a general background along the lines indicated by the title, but it is by no means a signal to start work immediately on any of the protective structures described. However, it was thought that such information should be made available to responsible civilian officials and civilian engineers, so they might consider methods, plans and especially procedures that can be followed in their respective localities should the situation change.

The first section of the pamphlet is devoted to the weapons, the ballistics and the effect of weapons used in aerial attack, while another section lists measures of defense against aerial attack, including protection of buildings, protection of utilities and industrial plants, and air raid shelters. In addition to discussion of the problem of design of air raid shelters, designs of several types of bomb resistant shelters are reproduced.

The section concerning weapons used also discusses the effect produced by demolition bombs, fragmentation bombs, gas bombs, incendiary bombs, armor piercing bombs, aerial mines, and aerial gunfire. An account is given of the bearing that trajectory, angle of impact and impact velocity have in an aerial attack. The general effect of bombs on buildings, utilities, railroads and bridges also is dealt with briefly. Special measures necessary for the protection of industrial plants, utilities and buildings are listed.

In the section describing air raid shelters, problems of design, general requirements, shelters in existing buildings and external shelters are discussed. Illustrations in the pamphlet include a demonstration of the proper method of using sandbag walls, features of buildings in industrial areas which affect the selection of locations for shelters, and plans for construction of shelters for six persons, 50 persons, 100 persons, and 200 persons.

The material was prepared after a close study of European information, and is intended as a resumé along general lines of the latest information available. As additional material becomes available and is digested, it is planned to disseminate it to the proper authorities.

APPOINTED TO CONSTRUCTION COMMITTEE

The war department announced last month the appointment of Richard Henry Tatlow III as a member of the construction advisory committee of the office of the quartermaster general to fill the vacancy created by the resignation of Francis Blossom. Since July, 1940, Mr. Tatlow has served as assistant to the chief of the engineering branch of the office of the quartermaster general and has been in charge of the investigation and analysis of architect-engineering firms and construction contractors by that office. Prior to July, 1940, Mr. Tatlow was a partner in the engineering firm of Harrington & Cortelyou of Kansas City, Mo. In 1932 Mr. Tatlow served as an assistant to the engineers' advisory board of the Reconstruction Finance Corp. and in 1933 as a consultant to the public roads administration.

Serving on the construction advisory committee, in addition to Mr. Tatlow, are Brigadier-general George R. Spalding, chairman, F. S. Harvey, F. J. C. Dresser and Alonzo Hammond.

RESEARCH FACILITIES EXPANDED

A $35,000 extension of laboratory facilities is in progress at Battelle Memorial Institute, Columbus, Ohio —the result of increased research for industry, occasioned partly by the defense program but mainly by new products research. Much of the new work is being sponsored by industries that are planning to surmount the adjustments of the post-emergency period with new and improved products and reduced production costs.

Eighteen new laboratories and offices, totaling nearly 10,000 sq ft, are being installed to take care of forty additional research engineers. The majority of these laboratories are in the new laboratory building which was completed only a year ago.

WHAT MARKS A PROFESSION?

What marks off the life of an individual as professional? First, I think we may say that it is a *type of activity* which is marked by high individual responsibility and which deals with problems on a distinctly intellectual plane. Second, we may say that it is a *motive of service*, as distinct from profit. Third, is the *motive of self-expression*, which implies a joy and pride in one's work and a self-imposed standard of workmanship—one's best. And fourth, is a *conscious recognition of social duty* to be accomplished, among other means, by guarding the standards and ideals of one's profession and advancing it in public understanding and esteem, by sharing advances in professional knowledge and by rendering gratuitous public service, in addition to that for ordinary compensation, as a return to society for special advantages of education and status.

Next, what are the attributes of a group of persons which mark off their corporate life as professional in character? I think we may place first a *body of knowledge* (science) *and of art* (skill), held as a common possession and to be extended by united effort. Next we may place *an educational process* of distinctive aims and standards, in ordering which the professional group has a recognized responsibility. Third in order is a *standard of qualifications*, based on character, training and competency, for admission to the professional group. Next follows a *standard of conduct* based on courtesy, honor and ethics to guide the practitioner in his relations with clients, colleagues and the public. Fifth, I should place a more or less formal *recognition of status* by one's colleagues or by the state, as a basis of good standing. And finally *an organization* of the professional group based on common interest and social duty, rather than economic monopoly.—WILLIAM E. WICKENDEN, president, Case School of Applied Science, in an address delivered before the annual banquet of the Engineering Institute of Canada.

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Published monthly by the Keeney Publishing Company, 6 N. Michigan Ave., Chicago. Telephone: State 6916/ F. P. Keeney, President; W. J. Osborn, Vice-president; R. Payne Wettstein, Secretary; Chas. E. Price, Treasurer/ Address all correspondence to Chicago office/ New York office, Room 1734, Grand Central Terminal Bldg. Telephone: Murray Hill 9-8293/ Cleveland, 2128 Rossmoor Rd., Cleveland Heights. Telephone: Yellowstone 1540/ Los Angeles, 1406 S. Grand Ave. Telephone: Richmond 6191/ Subscriptions: U. S. and possessions, Canada, Mexico, Central America, South America, $2.00 per year; other, $4.00. Current copies, U. S. and possessions, 25c each, back copies 50c each. January, 1941, issue, $1.00 per copy/ Member Audit Bureau of Circulations and Associated Business Papers, Inc./

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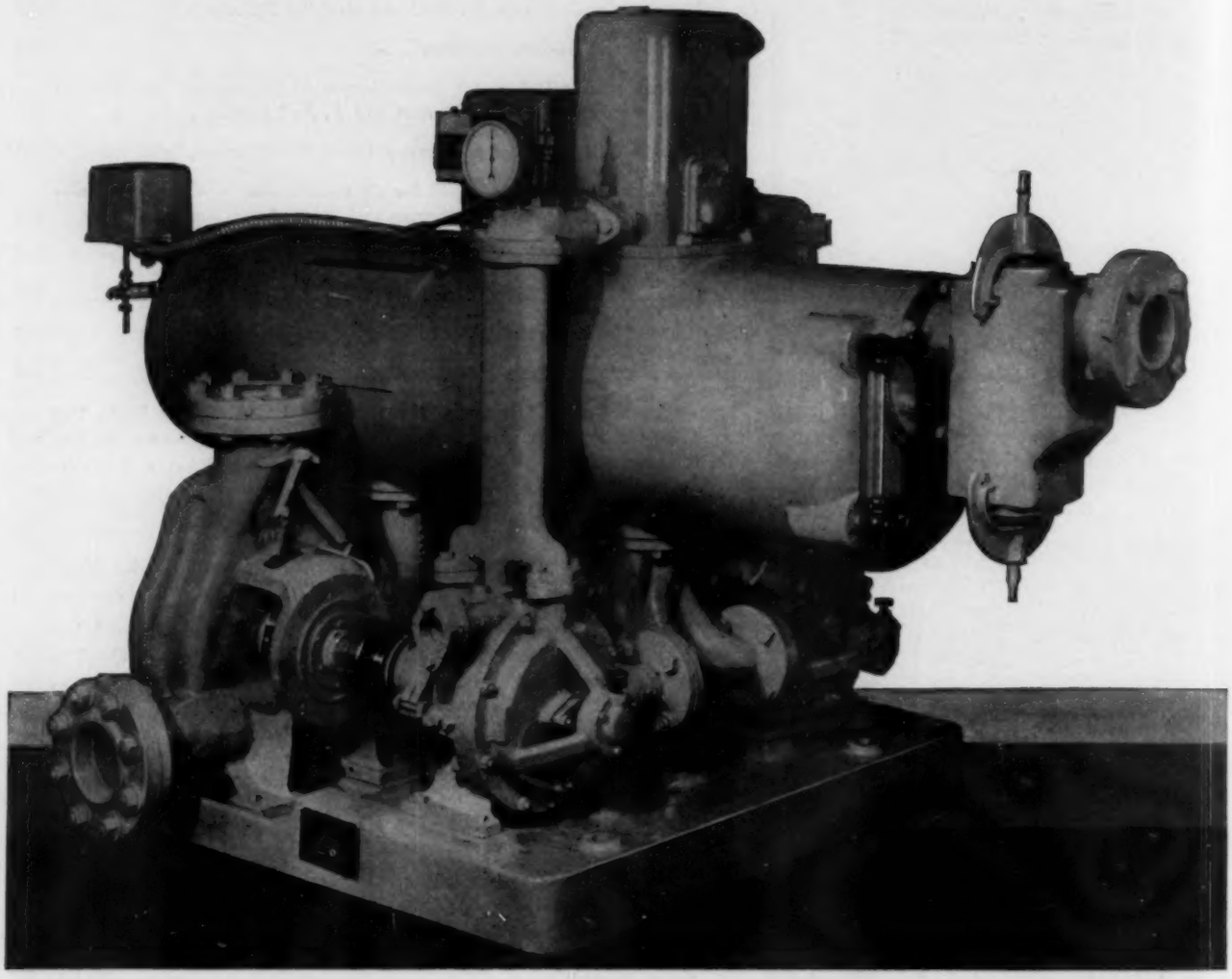
Where building heat *MUST NOT FAIL* always specify the Nash Vapor Turbine



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Construction for Defense

Central Steam Heating Plant for Army Camp Hospital

CENTRAL PLANT — Typical of many army camps being built under the national defense program is Camp Wheeler. Designed for a population of 19,000 men, it includes a 700 bed hospital group of 56 steam heated buildings served by a central boiler house, with steam distribution and return piping situated in the covered walkways. . . . . Steam is generated and distributed at 100 lb, is reduced at the buildings to 40 lb for unit heaters, sterilization, water heating, cooking, etc., and to 5 lb for direct radiation. . . . . Factors governing the design of the system were the temporary nature of the camp (expected life is 5 yr), the need for speedy construction (the steam system had to be completed in 60 days), the necessity for dependable steam service, and economy in installation and operation

Distribution Piping in Covered Walkways Is Standard Practice . . . By Frederick W. Bull* and H. King McCain†

TYPICAL of many new army camps being built under the national defense program is Camp Wheeler, at Macon, Ga. This cantonment, a replacement center for initial training of draftees, is designed for a populaton of 19,000 men and includes a 700 bed hospital group. In this group are 56 steam heated buildings scattered over a 65 acre tract, interconnected by covered walkways and served by one central steam plant. This involves a system of 8000 lineal feet of covered walks and three miles of distribution piping.

Although the construction division of the quartermaster general's office furnished stereotyped plans for each building, including a steam layout, it was necessary for the project architect-engineer to design the outside steam distribution system.

Coördination Essential

This problem involves the closest coördination of architectural and engineering planning on a project of this kind. For example, the buildings must be placed so that the hospital group will form a functional unit satisfactory to the Surgeon General. Next, the buildings and walks need to be oriented to allow easy access for service and fire protection. Building elevations must fit the terrain with minimum grading, and at the same time permit a slope in the covered walks not to exceed 5 per cent. (Liquids in food carts spill if the slope is steeper.) Too many low points must be avoided, since each low point requires a vacuum pump for condensate. Then, too, the walkways must be large enough and strong enough to house and support the steam piping.

Standard Army Design

The buildings themselves are of standard army "temporary housing" frame design, with wallboard inside and metal siding outside. Included in the group are ward buildings, clinics, mess halls, quarters for officers and nurses, barracks for enlisted personnel, recreation buildings, administration buildings, storehouses and a morgue. The boiler house has a structural steel frame with corrugated steel sides and roof. The interconnecting walkways are of frame construction with metal siding.

Heating is provided by direct cast iron radiation in sleeping rooms and clinics, and by suspended unit heaters in public rooms, such as mess halls.

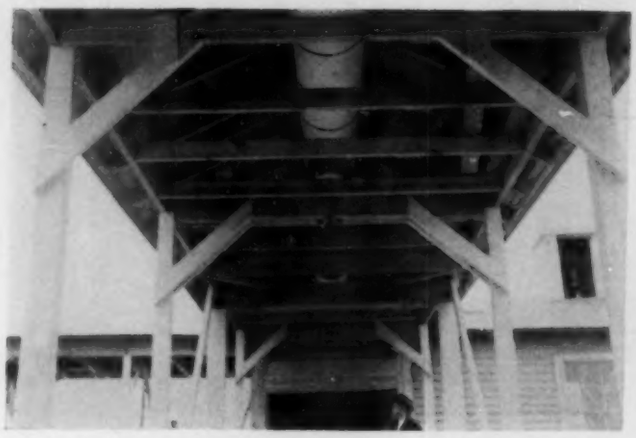
The direct radiation is manually controlled and the unit heaters have thermostatic control.

Steam is generated at the boiler house in two boilers of the three drum, bent tube, low head type. Each boiler is fired with a double retort, ram type, bituminous coal stoker having side dump plates. The two boilers, when operating at 200 per cent of rating, will jointly develop 1600 hp. Boiler room accessories include complete automatic control of stokers and breeching dampers; other standard accessories include feedwater pumps, water level control, feedwater heater, surge tank, soot blowers, tube cleaners, blow off tank and a portable coal conveyor. The steel stack serving the boilers is 90 in. in diameter and 115 ft high.

Mains Run in Walks

Steam for the plant is generated at a pressure of 100 lb saturated and is distributed to the buildings at this pressure. The steam mains are run in the covered walks connecting the

*Heating engineer, Hentz, Adler & Shutze with Newcomb & Boyd, architect-engineer, Camp Wheeler.
†Utilities engineer, Hentz, Adler & Shutze with Newcomb & Boyd, architect-engineer, Camp Wheeler.



***At left*—Overhead steam line on roller support. There is a valve in the cupola at right side of this view. . . . *At right*—Return and vacuum pump discharge lines under walks with footings for anchors in foreground. The vacuum pump line is covered, but return line covering had not been completed when photo was taken**

buildings and the pressure is reduced at the buildings to 40 lb for sterilization, unit heaters, water heating, cooking, etc., and to 5 lb for direct radiation.

The medium pressure returns are dripped through flash tanks and join the low pressure returns which run by gravity to vacuum pumps which are situated at the nine low points of the system. The discharges from these pumps are returned to the boiler house.

The steam mains in the covered walks are run above the lower chord of the roof truss where space conditions are limited, and the pitch of the pipes must follow the slope of the walks. At all points where changes in pitch occur the mains are anchored, and the elevations of walks and buildings are controlled so that these changes of pitch are always from a lesser to a greater slope in the direction of flow, to prevent the accumulation of condensate. The connections to the buildings are taken from the bottom of the mains to drip the lines.

Welding Facilitates Job

Electric arc welding of all distribution steam and return piping facilitated the making of elbows required by changes in pitch. The returns from the buildings are run under the floor of the walks to the pumps, and the discharges from the pumps flow under the walks to the feedwater heater in the boiler house.

Due to the restricted space in which the steam mains were run, it was necessary to provide cupolas in the roof framing for the installation of the main shut off valves. Drips are provided ahead of these valves and run to the nearest flash tank.

The expansion is cared for by joints of the double slip, internally and externally guided type. The alignment of the pipe at the joints is further insured by installing double guiding rolls on the pipe within 7 ft, and on each side, of the expansion joints.

Removable sections were provided in the roofs over the expansion joints to facilitate servicing.

How Piping Is Anchored

The framing at anchor points is designed to withstand a thrust of 20,000 lb on the 8 in. and 10 in. piping and 10,000 lb on the smaller lines. This is accomplished by carrying the 8 in. and 10 in. anchor points independently of the wooden walkways using structural steel supports resting on concrete "deadmen." The anchor points on the smaller lines employ structural steel reinforcement of the wooden members of the walks. The returns under the floor and the discharge lines from the vacuum pumps are anchored to concrete blocks set in the ground.

All distribution piping is covered with standard thickness fibrous glass insulation.

FLOODING ROOFS

In his article entitled Flooded Roof, Double Pane Windows Assist in the Control of Air Conditions in the April HPAC, M. C. Haley sets forth very clearly the reasons for ponding of the roof which, in this particular instance, will assist in the stabilizing of the heat transfer effect through the roof, but will also avoid conflict with local ordinances relating to storm drainage; providing, as it will, a reservoir for gradual disposal of storm water to the storm drainage system which, as stated, is not always adequate to meet the maximum experienced drainage demands.

It is not at all certain, in my opinion, that much heat resistance benefit will be derived from a 2 in. deep *still* water pond on a roof. The normally cool climatic conditions prevailing in the Oakland, Calif., area would not seem to indicate any great advantage to be expected except as a stabilizing factor. The accumulation of soot, dust and oily substances on the surface of the water will tend to retard evaporation. Consequently we should expect that the still water pond will be warmed many degrees above the surrounding air temperature by the solar heat effect during the daytime periods.

It is apparent that the author is cognizant of the probable limitations since it is stated in his article that provisions have been made to drain the roof of water, if desired.—E. P. Heckel, E. P. Heckel & Associates. Member of HPAC's board of consulting and contributing editors.

RELIEF PROGRAM *for* Hot Water Tanks

F. M. Dawson and A. A. Kalinske of Iowa Institute of Hydraulic Research Tell of Safeguards for Hot Water Systems

HOT WATER is often furnished for industrial or domestic purposes from some type of storage tank in which the water may be heated directly or may be supplied to it from an external heater. The main purpose of such a storage tank is, of course, to permit taking care of momentary high hot water demands, even though only a low input heater is used.

In order to secure the full advantages of a storage tank system for supplying hot water, careful attention should be given to the piping connections and other installation features. It is unusual for the tank to be completely full of hot water. Instead, the top part may contain hot water while the lower part may have entirely cold water. If proper service is to be provided, it is necessary that the hot water remain undisturbed and not mixed up with the colder water; otherwise, it may be necessary to heat the entire content of a tank before a sufficient amount of hot water is available. In other words, the heater should place the heated water in the top part of the tank and the piping connections to the tank should make possible the drawing off of hot water from the tank top and the supplying of cold water to the tank bottom without any great disturbance. The difference in densities between water heated to approximately 150 F and cold water at approximately 50 F, or thereabouts, is sufficient to keep them separated unless violent flow disturbances occur.

Though a properly installed storage tank system has many advantages, there are certain hazards associated with the storing of a large quantity of hot water under pressure such as exists in the water supply system. It is the purpose here to discuss these hazards and to indicate methods of protection against the possibility of tank rupture and so-called hot water tank explosions.

Hazards in Heating Water

An ordinary storage tank is under a certain water pressure, depending upon what the static pressure happens to be in the system. When the water in the tank is heated by circulation through an external heater or by heating units in the tank, it gradually expands like most other substances when subjected to an increase in temperature. If the tank is connected to some supply, without any intervening check valve or obstruction, the increase in volume will cause part of the water to be pushed back into the supply line. If directly connected through the cold water service line to the city water main, the excess volume due to expansion simply backs up into the cold water line. If the heating is sufficient, the backing up may be so extensive as to cause hot water to reach the water meter. This may ruin the composition discs in the meter.

RELIEF PROGRAM—A great deal of research work has been done at the Iowa Institute of Hydraulic Research on safeguarding industrial and domestic hot water supply systems. The authors explain here the reasons for so-called "explosions" of hot water supply tanks, which are due to the sudden release of pressure on superheated water. A certain part of the water will flash into steam and as an equivalent weight of steam has a tremendous volume as compared to water, there is a large energy release involved. . . . The authors discuss the required capacities of pressure and temperature relief devices for such tanks, with recommendations for installation. Formulas are presented for required discharge capacities of relief devices

If the entire content of a 500 gal storage tank is heated from 70 F to 160 F, there will be an increase in volume of about 8 gal. If the cold water line is of 2 in. size, this 8 gal will fill about 45 ft of pipe. Of course, this does not mean that hot water would go back 45 ft, because the first part of the back flow might be cold water.

If, for any reason, the above mentioned back flow cannot occur (due to a check valve or a pressure reducing valve in the line, for instance, or because of temporary shut off of the cold water line) the pressure in the tank will rise as heating continues. Obviously, such a pressure may reach considerable proportions if the heating continues for any length of time. Rupture of the tank seams could readily occur under such conditions with the possibility of flooding the building. Nothing else would happen if the water temperature at the time of tank rupture was less than 212 F, the atmospheric boiling point. No so-called "explosion" could result.

If the tank water is not confined, continued heating would of course cause no increase in pressure beyond ordinary line pressure. However, the temperature would continually increase. For example, if the water were under a city pressure of 75 lb ga, it could reach a temperature of 320 F without any steam forming inside the tank. If tank failure should result under such conditions, then the sudden release of pressure would cause part of the superheated water to flash into steam. If the water in a 500 gal tank is heated to 320 F, the sudden release of pressure would cause about 10 per cent of the water to flash into steam. Since a given weight of steam has a volume almost 1700 times as great as the

same weight of water, some 82,000 gal of steam at atmospheric pressure would form. The energy released under such conditions is equivalent to that produced by the explosion of about 15 lb of nitroglycerine. Naturally, tremendous damage and loss of life could result if this occurred in any building.

The failure of the storage tank at times when the water is superheated may occur due to excessive pressure if the tank is not connected to an open system, or due to weakening of the tank seams by corrosion, further aggravated by the strains introduced by the excessive temperature. Also, a tank may rupture due to the collapse caused by creation of a partial vacuum. This partial vacuum may be the result of a sudden drop in pressure because of the flow of a large amount of water from the street main—such as may occur during a fire or when the main breaks.

Pressure Relief Devices

In order to guard against development of excessive pressure inside a hot water storage tank due to thermal expansion of the water, it is customary to install a spring loaded pressure relief valve set to open at a pressure about 20 lb higher than the normal line pressure. Even though the storage tank is connected to an open system, such a pressure relief device is desirable, since somebody might shut off the cold water line to the tank without making certain that the heater does not operate. This device should be installed in the cold water line near the tank and there should not be any valves between it and the tank. The drain or drip from the relief valve is carried to some point over a sink or floor drain. *It should never be connected directly into any drain, waste or vent pipe.*

The amount of water discharged by any pressure relief device is quite small, since it takes only a small quantity of water to relieve any pressure due to the thermal expansion of the water at a rate that will limit the pressure rise for any given heat input to 10 per cent of the pressure at which the valve is set to open. This discharging capacity can be easily evaluated by a simple test. For any given installation it should exceed the value obtained from:

$$Q = \frac{B\,(V_h - V_c)}{T_h - T_c} \quad \ldots\ldots\ldots\ldots [1]$$

where Q = Gal per hr of water to be discharged;
B = Heat input of heater in Btu per hr;
V_h = Gal per lb of water at hot temperature, T_h;
V_c = Gal per lb of water at cold temperature, T_c.

The value of Q for any given value of B will tend to vary with the hot and cold water temperatures. From a curve indicating the thermal expansion of water, it is found that the maximum value of $(V_h - V_c)/(T_h - T_c)$ is about 5×10^{-5}. Thus [1] becomes:

$$Q = 5 \times 10^{-5} B \quad \ldots\ldots\ldots\ldots [2]$$

[*Note: Multiplying* by 10^{-5} means the same as *dividing* by 10^5, or by 100,000.]

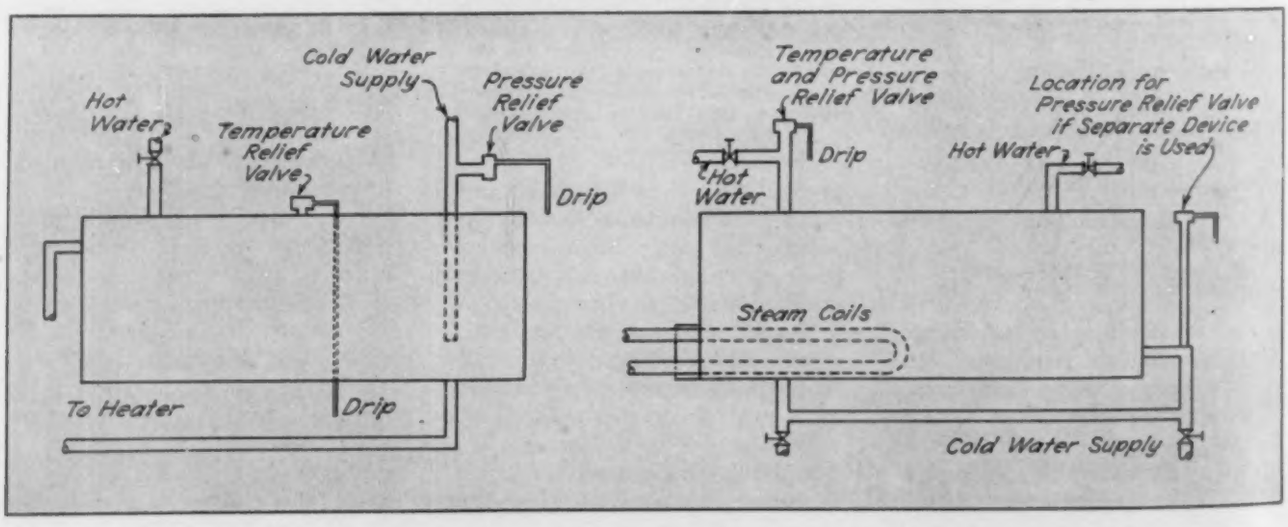
Data as to the value of Q for a pressure 10 per cent greater than that for which the relief device is set would permit a determination as to whether or not a device is adequate for any particular installation. The maximum value of heat input B for which a device can be used should be stated by the manufacturer.

Temperature Relief Devices

From the previous discussion it is obvious that a so-called hot water tank "explosion" can occur only if the water is heated to a temperature above the temperature at which boiling would occur under the atmospheric pressure. The tank must rupture first and thus suddenly relieve the pressure on the superheated water. The explosion or the sudden flashing of the water into steam occurs *after* the tank ruptures. Since the real cause of the explosion is the fact that the water was overheated, the preventive measure should consist of temperature control or safeguarding against the creation of excess temperature.

Two general methods of prevention of high temperatures are (1) the elimination of the source of heat, and (2) the admittance of cold water into the tank and the discharge of hot water after a predetermined temperature is reached. The first method is of course used on all types of automatic heaters and should be used whenever possible. However, absolute dependence should not be placed on such automatic control of the heat source. There is always the possibility of the control apparatus failing to function properly. Therefore, hot water tanks should be equipped with a suitable temperature relief device.

The authors' recommendations for installation of pressure and temperature relief valves on hot water storage tanks



One type of such a device has a fusible plug of metal which melts at about 212 F. The hot water then runs from the opening until the device is serviced. The disadvantages of this type of device are that it requires servicing after each operation, and also the temperature at which the metal plugs melt varies and this temperature may be affected by age and deterioration of the plug.

Another type of relief device is one which incorporates a bellows or bimetal disc, which opens at a temperature of 210 F and closes when the temperature drops to approximately 160 F. The operation of the valve is automatic; however, there is the possibility of sticking, and therefore such devices should be of first class construction and should be examined periodically. The operation of temperature relief devices should not depend upon the water pressure, since the pressure may be off. They should be activated by temperature.

Temperature relief devices should be installed at the point of maximum water temperature, which is usually in the pipe just above the heater or in the very top of the tank. Various suggested locations for a temperature relief device, pressure relief device or a combination device are shown in the figure.

Temperature relief devices should be rated as to their Btu capacity. That is, when the device opens at 210 F, it should be able to discharge sufficient hot water to prevent any further rise in temperature. The discharge capacity of the device at a temperature of 210 F and some standard water pressure, such as 30 lb, can readily be determined. The device should not be used on installations where this discharge capacity is less than the value obtained from the following equation:

$$Q = \frac{B}{8.34\ (210 - T_s)} \quad \text{.....[3]}$$

The cold water temperature is usually taken as 60 F; therefore, [3] becomes:

$$Q = 8 \times 10^{-4} B \quad \text{.....[4]}$$

One other item worthy of mention regarding temperature relief devices is to consider what would happen if the water pressure were shut off and thus no water were to flow from the device when it opened at a temperature of 210 F, because of no pressure to cause it to flow. Under such conditions, steam would be generated and would escape through the valve. Unless the opening through the valve were quite large, a pressure would build up in the tank with a resultant increase in temperature. This is of course not very desirable. It is therefore suggested that temperature relief devices have a Btu rating considerably in excess of the heater capacity.

How Air Conditioning Serves Industry

■ Precision grinding spindles turning at the rate of 50,000 rpm are being manufactured at the Landis Tool Co., Waynesboro, Pa., with the help of air conditioning. Operating at such high speeds, these spindles must be absolutely true, perfectly balanced and free from any vibration. It was discovered in the laboratory that metal parts for these high speed spindles—due to expansion and contraction caused by atmospheric variations—could only be assembled with perfect accuracy in an air conditioned room. Two air conditioning units, which maintain a uniform temperature of 70 deg and a dry and dust free atmosphere, were installed in a specially constructed assembly room at the plant. The room is carefully insulated, with double walls, windows and doors keeping it sealed tight from outside influences. Only filtered, conditioned air is admitted through the two units. The spindles assembled in this air conditioned laboratory carry the grinding wheel on a specialized type of machine used by the ball bearing industry.

■ The tendency for gases to rise makes the job of crane cab operators hazardous where gases are present in any appreciable degree. The confined position of the operator adds to his susceptibility to injurious effects from dangerous industrial gases and odors. In the Utah plant of one of the country's largest smelting companies an analysis of the air showed a sulphur dioxide content of 25 parts per million. To overcome this condition, gas adsorption equipment was recently installed in the crane cabs of the smelter to safeguard the operators' health. The air for ventilating the cab is drawn in through a field of highly activated coconut shell carbon which frees it of sulphur dioxide.

■ Dirt and dust particles, so small they are invisible to the naked eye, are still large enough to make what otherwise would be a good television tube, bad. In the laboratories of the Farnsworth Radio and Television Corp., Ft. Wayne, Ind., it was found that these minute dust particles caused a spotting of the cathode of television dissector tubes during their treatment with photo sensitive material. Installation of a 6000 cfm electrostatic air filter to clean the laboratory air of the invisible dust and dirt has eliminated the tube cathode spotting, permitting perfect tubes to be made consistently. The dissector tube is the heart of the television process; it is the tube that picks up the image, dissects it into separate elements and converts each image element into transmittable electrical impulses. Its manufacture is more delicate than that of a fine watch, for electronically it must be almost perfect. Prime requisite is a cathode whose emission must be uniform.

■ A shortage of labor in America's booming diamond cutting industry may be solved as a result of the first air conditioning installation in a New York City diamond sawing room. Excessive humidity has been one of the major obstacles in diamond cutting; a diamond saw won't work in wet, muggy weather and even slight excess humidity slows up the process of sawing the precious stones. In an effort to eliminate this condition, an air conditioning unit was installed in the diamond sawing room at Baumgold Bros., where it maintains a definite humidity and temperature. As a result of air conditioning the room, the diamond cutters are able to keep their saws running continuously. This has increased their sawing capacity by 30 per cent.

MODEL PLANT

Exhaust Steam Used for Air Conditioning and Heating at United States Tobacco Co.'s Plant. By G. C. Dittman and O. M. Schneider, Schmidt, Garden & Erikson, Architects and Engineers

ON THE outskirts of historic Richmond, Va., the United States Tobacco Co.—makers of "Model," "Dills Best," "Old Briar" and "Tweed" smoking tobaccos—recently completed a new processing plant. Air conditioning plays an important part in the processing of tobacco, and is of course incorporated.

Six stories high and 100 ft by 350 ft in area, the plant is modern in design and houses offices, shipping room and all other necessary departments. The power house is a separate building approximately 200 ft from the plant and connected to it through an underground tunnel in which steam, air, vacuum and electric services are carried.

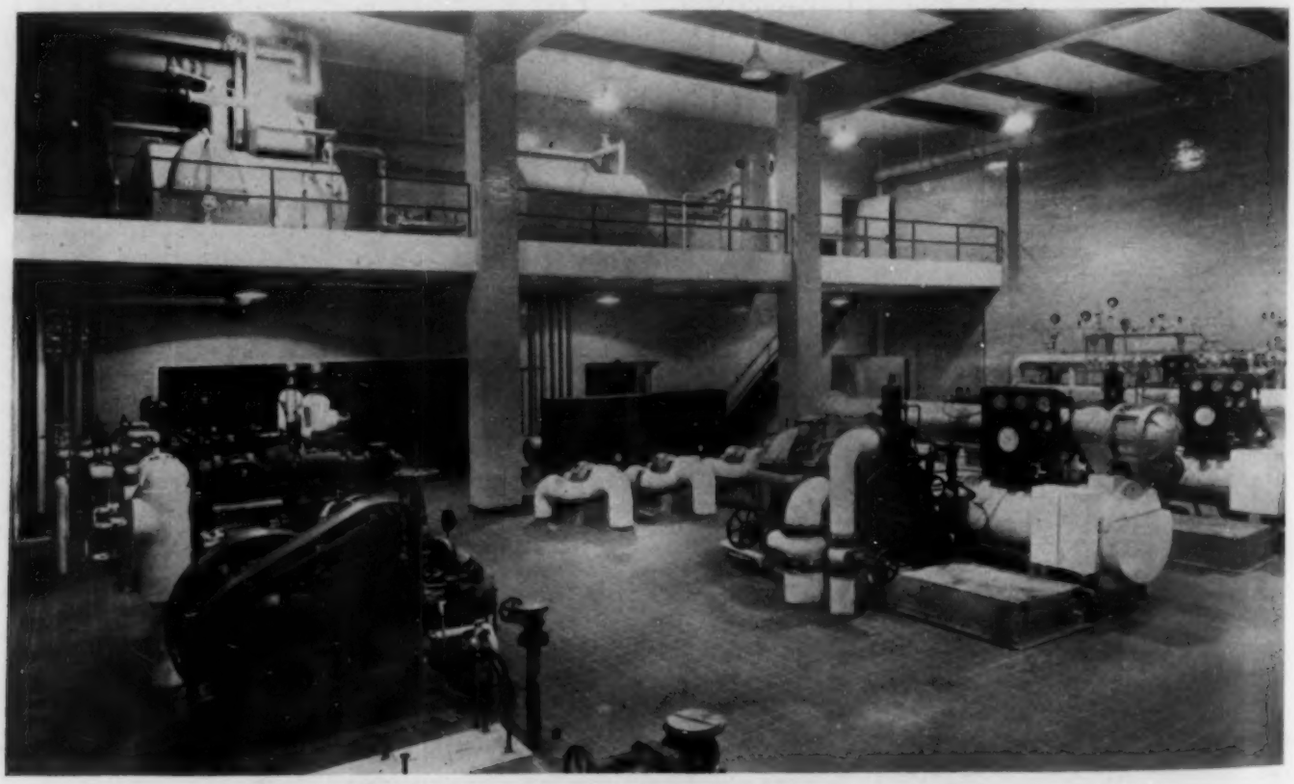
Probably the most interesting feature of the air conditioning design is the use of exhaust steam for refrigeration for air conditioning, and for heating in winter. There are two 150 ton centrifugal refrigerating machines in the power house which furnish all the chilled water required for the air conditioning systems. These units operate at 7640 rpm and are directly connected to 139 hp steam turbines, as indicated in the flow diagram.

In order to take care of the electrical needs of the plant, two 400 kw engine-generator sets were provided. The engines operate at 175 lb pressure, 500 F total temperature and exhaust at 5 lb pressure or operate condensing. During the winter, one engine supplies the steam for heating while the second engine operates condensing; or, if necessary, both engines can operate to supply 5 lb steam to take care of the heating load.

Condensing Water

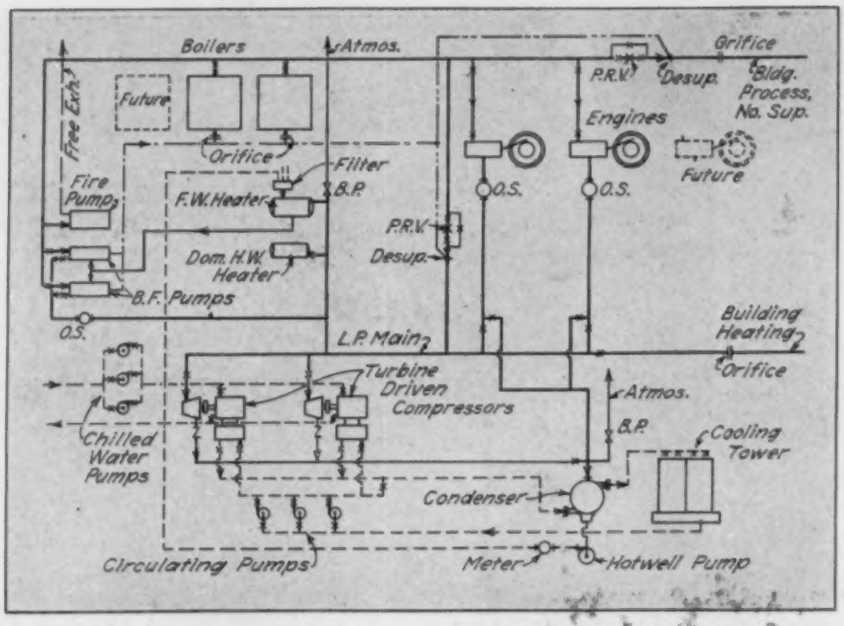
During the summer, both engines operate at 5 lb back pressure, which is the steam operating pressure of the turbines driving the refrigeration machines. The turbine units exhaust at 26 in. vacuum into a condenser. This same condenser is used by the engines in the winter and by the refrigerating machine turbines in the summer.

For supplying cooling water to both the steam condenser and the



The pressure reducing and desuperheating station for process steam is on the wall at the extreme right of this picture. In the right foreground are the two turbine driven centrifugal refrigerating machines. On the mezzanine floor at the extreme left is the filter for filtering the condensate before it goes to the feedwater heater to the right of the filter. The feedwater heater is of the open deaerating type. The capacity of the feedwater heater is 60,000 lb per hr

MODEL PLANT — The United States Tobacco Co.'s new processing plant — where "Model," "Dills Best," "Old Briar" and "Tweed" smoking tobaccos are made —is noteworthy for its modern design and mechanical features. Two 150 ton centrifugal refrigeration machines driven by steam turbines chill water for air conditioning which is piped to the air treating apparatus in the main building through an underground tunnel connecting it and the separate power house. Steam at 5 lb is furnished these turbines in summer from the engines driving the electrical generators, the turbines exhausting at 26 in. vacuum into a condenser. In winter, one of the engines supplies 5 lb steam for building heating and the other operates condensing. . . . The same condenser is used by the engine in winter and by the refrigerating machine turbines in summer, and there is a cooling tower for supplying water to both the steam and refrigerating condensers. . . . The excellent photograph above was taken in the power house; it's quite a peaceful scene, isn't it?



Exhaust steam from the engines driving the generators goes to the turbines driving the centrifugal compressors in summer, the turbines exhausting at 26 in. vacuum into the condenser. In winter, one of the engines supplies steam for heating the plant, and the other operates condensing. The same condenser is used by the turbines in summer and the engine in winter. (O.S. = oil separator, B.P. = back pressure valve, and P.R.V. = pressure reducing valve on diagram)

refrigerating condensers, a cooling tower was installed on the roof of the engine room. It has capacity to cool 1250 gpm from 108 F to 85 F with an outdoor temperature of 95 F dry bulb and 80 F wet bulb. The tower is of the wood filled, forced draft type and is provided with two variable pitch propeller fans.

During the winter, the condensing water will go from the cooling tower to the condenser and then directly back to the cooling tower. During the summer months, however, the circulating water leaves the cooling tower at 85 F and first passes through the refrigerating machine condensers, leaving at 92 F. Then it passes through the steam condenser, leaving at 108 F, and goes to the cooling tower.

Three condenser circulating water and three chilled water circulating pumps, each having a capacity of 500 gpm, are installed. All these pumps are motor driven and are manually controlled.

Control of Turbines

Of particular interest is the control on the turbines driving the refrigerating units. Each turbine is provided with a constant speed governor which will hold the turbine speed to 7640 rpm, the normal rating of the machine. In case of emergency, an adjustment can be made whereby the capacity of the machine can be increased 10 per cent. An emergency trip governor is also provided for safety.

In addition to the above mentioned governors, there is a direct acting variable speed hydraulic governor which controls the speed of the machines at loads below normal. This governor is controlled by a thermometer bulb inserted in the chilled water return line and is so designed to maintain a constant chilled water outlet temperature regardless of load. At normal rating, the chilled water temperature is 42 F and the return water temperature is 48 F.

Close-up view of the three condenser water circulating pumps and the three chilled water circulating pumps. In the background is the remote pushbutton panel board for controlling the motors

Several Conditioning Systems

The chilled water is circulated to the air conditioning apparatus in the main building through piping installed in the underground service tunnel.

The second and third floors of the factory area, as well as the offices and employees' dining room, are all air conditioned. The third floor of the factory is devoted to bulk storage of the tobacco, while the second floor is the packaging department. Air conditioning these areas insures the consumer getting his tobacco in the best condition. The air conditioning installation consists of a number of separate systems, which —for the purpose of simplification —are described by numbers:

Supply system S-1A, and the accompanying exhaust system E-1A, furnish 5060 cfm to the first floor office areas. The offices are maintained at 75 F dry bulb and 58 per cent relative humidity in summer, and 70 F dry bulb and 50 per cent relative humidity in winter.

Supply system S-1B, and its exhaust system E-1B, serve the dining room, matron's office and the first aid room on the second floor. These areas are maintained at 75 F dry bulb and 58 per cent relative humidity in summer and 70 F dry bulb and 50 per cent relative humidity in winter.

The air treating apparatus for systems S-1A and S-1B is in the basement.

System S-2, and its exhaust E-2, supply 41,100 cfm to the second floor factory area with air at a temperature of 75 F dry bulb and 52 per cent relative humidity in summer and 75 F dry bulb and 52 per cent relative humidity in winter.

System S-3, and its exhaust fan E-3, furnish 39,300 cfm to the third

floor factory area with air at a temperature of 75 F dry bulb and 63 per cent relative humidity in summer and 75 F dry bulb and 63 per cent relative humidity in winter.

Systems S-2 and S-3 are supplied from dehumidifier units situated in the pent house on the roof. These are spray type dehumidifiers supplied with chilled water in the summer and hot water in the winter by an ejector water heater for the sprays. Steam coils are used for preheater and reheater coils. The dehumidifiers are constructed of 18-8 stainless steel to preclude the possibility of corrosion.

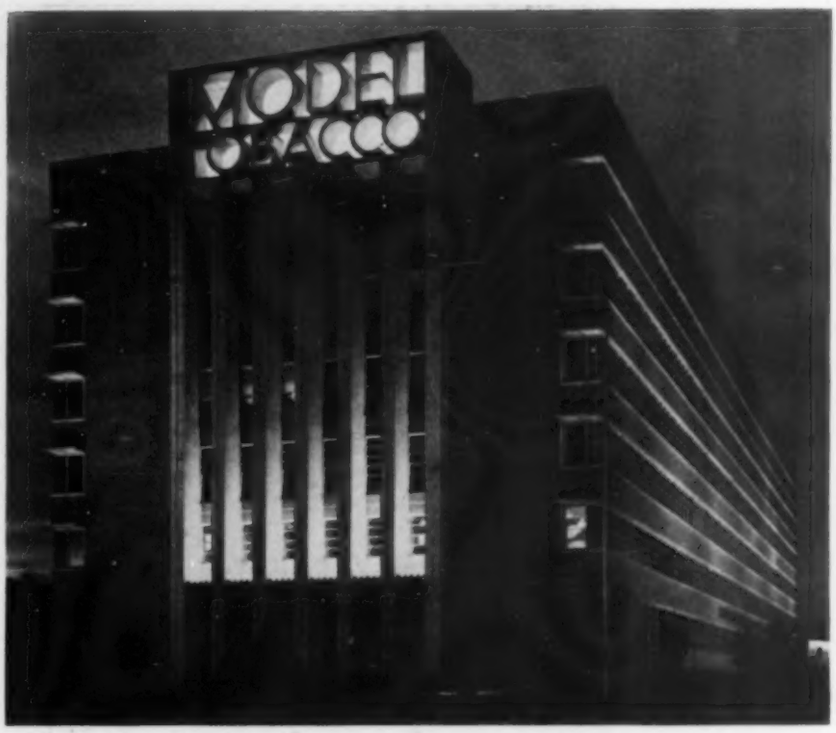
Supply system S-4, also situated in the pent house, supplies heated air only for the toilet and locker rooms on the second floor. All the units are designed to handle 25 per cent outdoor air and 75 per cent recirculated air.

Systems S-1A and S-1B situated in the basement machine room, have intake and exhaust louvers in the building walls above grade in the basement. Systems S-2, S-3 and S-4 and their corresponding exhaust systems have supply louvers in the pent house walls and discharge goosenecks extending above the pent house roof.

All of the duct work in the factory space is exposed, while the ducts in the office area and dining room area are concealed above suspended plaster ceilings. The offices and dining room, matron's office and first aid room have high velocity ceiling type supply outlets. The factory area has side wall deflecting registers mounted on the side of the duct runs.

Two large tobacco dryers are installed on the sixth floor of the building. When these are in operation they each require 50,000 cfm of 70 F air which, after passing through the dryers, is discharged above the roof to atmosphere. To supply this air eight heating units are suspended at the ceiling. Each unit has an individual outside air intake. These units are controlled in groups of four by two room type thermostats, with pilot duct stats in the fan discharges. These duct stats in turn control diaphragm valves on the steam coils and face and bypass dampers.

The control system is of the pneumatic type. For the two large supply systems S-2 and S-3, during summer



Noteworthy for its modern design is the new home of "Model" tobacco

operation a dew point thermostat controls a three way valve on the chilled and recirculated water lines and the face and bypass dampers in the dehumidifier units. A thermostat placed in the dehumidifier fan discharge controls the diaphragm valves on the steam supply to the reheater coils.

In the winter, the dew point thermostat controls the maximum outside air damper, return air damper and the diaphragm valves on the heating coils and pilots a quick acting, non-freeze thermostat which controls diaphragm valves on the preheater coils when the temperature drops below 35 F.

On both the second and third floor, there are control panels in the conditioned space with adjusting and recording mechanisms for all control of the two units so that it is not necessary for the man in charge of the floor to go to the pent house in order to change the setting of any of the control instruments.

For the office section, booster heaters are controlled by room thermostats to obtain the desired conditions. There are volume control dampers with minimum positions governed by room type thermostats for the dining room and offices on the second floor.

Heating of the factory space is by means of unit heaters operating on a vacuum system. These are the sole means of heat on the floors that do not have conditioning and they take care of the wall and glass losses on conditioned floors.

These units are controlled by thermostats adjacent to each heater in the daytime. At night, however, in order to maintain sufficient heat to keep things from freezing, but not sufficient to maintain 70 F as in the daytime, they are on a night control system. Each floor has a night thermostat in the factory with transfer switches to change over from day to night operation. These night thermostats have a range of 38 to 70 F. There is a master switch for each floor in the pump room of the power house so that the attendant on duty there can control the unit heaters at that point. Indicating lights are furnished in conjunction with the transfer switches to inform the operator of the control for each floor. Direct radiation which is installed in the basement is also under temperature control.

Considerable credit for the successful operation of the entire air conditioning system, including its numerous special applications, is due to T. A. Clark, chief engineer of the United States Tobacco Co., who worked very closely with Schmidt, Garden & Erikson, architects and engineers, in the design.

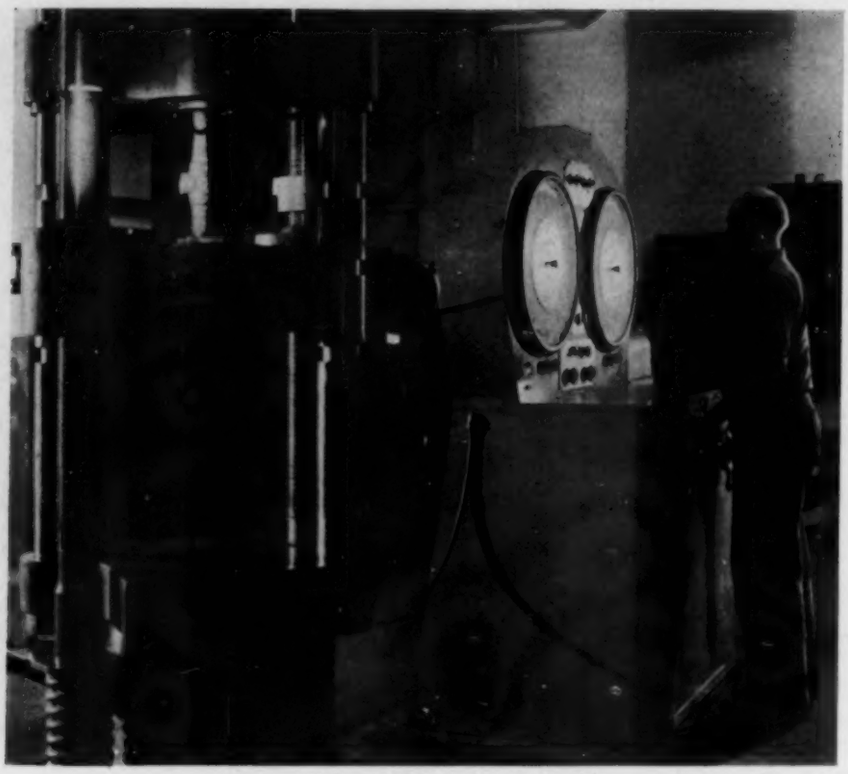


Fig. 1 — A welded assembly consisting of pipe and a tee being tested on a hydraulic tensile testing machine

STEELS for Welding Fittings

J. J. Kanter* Discusses New ASTM Spec

THE problems presented in developing a materials specification for factory made wrought carbon steel and carbon molybdenum steel welding fittings were outlined in my article in the April HPAC. Information on the various processes used in the manufacture of such fittings was included, as was a discussion of the committee organization and the scope of the specification, which is designated as A234-40T and published as tentative by the American Society for Testing Materials, 260 S. Broad St., Philadelphia, Pa.

The sections of the specification pertaining to chemical requirements, heat treatment, highly alloyed steels, and manufacturing operations were also considered in April. This month we take up carbon molybdenum fittings, welding operations, testing and certifying, and the marking of fittings.

*Chairman, section on welding fittings of subcommittee XXII, ASTM committee A-1 on steel, and research metallurgist, Crane Co.

STEELS — The usefulness of the new ASTM tentative specification for welding fittings, A234-40T, may not be fully apparent at first glance to the purchaser of welding fittings. The purchaser may perhaps be puzzled as to the distinct purpose for which such a materials specification is needed. The question may be raised as to why simple reference in ASA B16.9 or B16.11, the dimensional standards, to the approved ASTM standards for pipe, plate and billet is not sufficient. Why must this new number A234-40T be carried along in with all of the others? . . . The simplest and briefest reply to this question is that A234-40T covers the quality of the finished product and permits the purchaser to take advantage of the various fabrication methods. The type and form of the material with which the manufacturer starts are so varied that a specification so cast enables the user to take advantage of the available material with a minimum of different specifications. Thus, by using A234-40T for the material designation for wrought welding fittings, the purchaser can dispense with choosing among the detailed requirements of a number of ASTM specifications

Carbon Moly Fittings

Carbon molybdenum steel fittings are required by A234-40T to be heat treated according to the ASTM prescriptions for the type of material from which they are fabricated. In the case of pipe covered by A206-40T, whether hot or cold finished, a stress relief anneal is required. The requirement for heat treatment under A204-39 for fittings fabricated from molybdenum steel plate are less exacting than those for carbon molybdenum steel pipe. No heat treatment definitions are stated in A204-39, other than direc-

tions that heavy plate over 2 in. in thickness shall be uniformly heat treated to produce grain refinement. Therefore, the reference in A234-40T to the heat treatment prescription of A204-39 (see Table I, reprinted on p. 214, April HPAC, by permission) may be open to some question in view of the fact that welding fittings, where fabricated from plate, will in general be made from lighter than 2 in. thickness.

Carbon molybdenum steel fittings forged from billets or bars, as for instance tees or reducers, are required to be heat treated under the provisions of the specification for alloy steel pipe flanges, A182-40, in which a heat treatment proper to design and chemical composition is specified, but the method of heat treatment is left to the option of the manufacturer. In view of the fact that carbon molybdenum steel used in fittings and governed by A182-40 is probably all fabricated by hot forming and hot finishing operations, the only questionable loophole in this part of the specification would be a manufacturer exercising the option to omit a separate stress relief heating on the grounds that grade F1 hot finished is suitably stress relieved. Strictly speaking, the requirement of A234-40T with respect to heat treatment of A206-40T fabricated pipe is more exacting than for forgings made from A182-40 grade F1 or plate fabrications using A204-39. This is a matter which probably will have to be clarified by the ASTM committee before A234 is finally adopted as a standard. It may develop through the tentative use of A234-40T that these requirements will need amplification to cover independently the heat treatment requirements for carbon molybdenum steel welding fittings.

Welding Operations

Fittings manufactured through the use of a welding operation, whether it be forge or fusion welding, are all required to receive a heat treatment subsequent to forming. In the case of a forge welding, a method not extensively used, either normalizing or full annealing after welding is prescribed in all cases by A234-40T because of the tendency for such an operation to leave the steel in a grain coarsened or "burnt" condition, if not followed by a proper heat treatment.

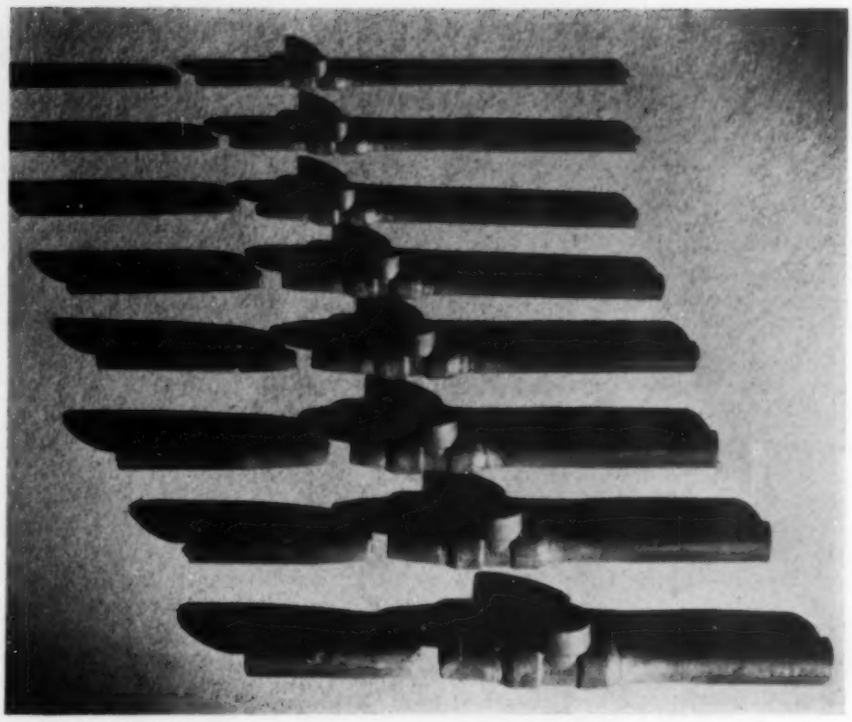


Fig. 2 — A group of typical assemblies, made with welding tees, subjected to a tensile pull test

Fusion welded fittings are required by A234-40T to be heat treated in accordance with code rules governing fabrication welding (ASA B31.1, section 6) in which the welding procedure, preheating practice and stress relieving treatment are prescribed. The latest version of the rules governing stress relieving, which will appear in the revised code for pressure piping, are as follows:

(a) (1) Welded joints in carbon steel shall be stress relieved by one of the methods described in this section when the pipe wall thickness is ¾ in. or greater. (2) Welded joints in carbon steel having a carbon content in excess of 0.35 per cent and in carbon molybdenum steel, shall be stress relieved when the wall thickness is ½ in. or greater. (3) In all cases, the member having the highest carbon content or the greatest wall thickness adjacent to the point of welding shall govern. When slip on type flanges are attached by fillet welds, the thickness of the pipe shall govern.

(b) Stress relieving shall be done by heating to a temperature between 1100 F and 1250 F. The parts heated shall be brought slowly to the required temperature and held at that temperature for a period of time proportioned on the basis of at least one hour per inch of pipe wall thickness for carbon steel and two hours per inch of pipe wall thickness for carbon molybdenum steel, but in no case less than one-half hour, and shall be allowed to cool uniformly.

(c) Stress relieving shall be accomplished by one of the following methods:—

1. Heating the complete structure as a unit.

2. Heating a complete section containing the weld or welds to be stress relieved before attachment to other sections of work.

3. Heating a part of the work by heating slowly a circumferential band containing the weld at the center. The width of the band which is heated to 1100 F to 1250 F shall be at least equal to twice the width of the weld* and the temperature shall be allowed to diminish gradually outward from the ends of the band.

*Width of weld is the width of the widest part of the welding groove. In no case shall the width of the band be less than the width of the weld reinforcement.

Testing and Certifying

Tension tests of the finished fittings are not required under A234-40T. In general, hydrostatic tests are a much better criterion of the quality and serviceability of fittings than are tension tests and provision is made in A234-40T that fittings be capable of withstanding a hydrostatic pressure test. Hydrostatic tests or tension tests for acceptance purposes are made a matter for agreement between manufacturer and purchaser, certification that finished fittings conform to the re-

quirements of A234-40T being the understood basis of acceptance. Large purchasers of fittings may wish to agree upon the definition of a "lot" of fittings and then decide upon suitable acceptance tests, provided it is possible to follow the fittings through the course of their manufacture. Where fittings are to be taken from stock, however, it becomes virtually impossible to identify fittings with their individual pedigrees and certification by the manufacturer that the fittings conform to the various mandatory requirements of ASA B16.9 or B16.11 and were made from the implied ASTM specifications seems to be the only workable basis of certification.

Chemical analysis upon the original material from which a lot of fittings has been made, of course, establishes the composition of the finished fitting as well as a check analysis upon the finished fitting. Certification of chemical analysis from the mill analyses should suffice to satisfy the buyer of the grade of steel, unless such identification either cannot be made or is not acceptable. A tensile test, however, made upon the original material prior to fabrication is not pertinent to establishing the properties of the finished fitting. The forming operations invariably alter the tensile properties and, in general, heat treatments are used on the finished product which establish different tensile properties than are to be found in the raw material. Thus, whenever any serious question arises as to whether the unprocessed steel has sufficient quality to yield the desired tensile properties in the finished product, tests must be made upon specimens cut from fittings. Where this is not possible, due either to small size, or shape, or expense of the fittings, demonstration of the quality of the steel may be resorted to by processing suitable samples for tension tests in such a manner as to obtain the benefit of the reduction and heat treatment accorded the actual fittings. Such tensile testing, however, is a matter for agreement between manufacturer and purchaser to suit a particular need, but becomes too involved as a basis for acceptance in standard specifications.

Marking Fittings

The marking requirements for wrought welding fittings set up in A234-40T are essentially those to be found in ASA B16.9 or B16.11. Some difficulty was encountered by the ASTM committee when considering marking requirements in connection with fittings on which steel stamp marking might be considered by an inspector to encroach upon the minimum wall thickness requirement for the fitting. In the case of alloy steel fittings, particularly, stamp marks are regarded by some as a possible source of cracks. A footnote in the specification advises that where steel stamps are used, care should be taken so that the marking is not deep enough to cause cracks or to reduce the wall thickness of the fitting below the minimum allowed.

The method of marking has been left open so that manufacturers preferring to use stenciling or tagging are permitted to do so. In the case of stenciling, or the use of decalcomanias, gummed labels, etc., of course, the chances that the marking will become obliterated while the fittings lie in stock is to be reckoned with. Some manufacturers prefer to identify small carbon molybdenum fittings with steel tags attached by spot welding. Again, however, the objection has been raised that the spot or tack welding necessary for the attachment of steel tags to an alloy steel fitting is quite likely to leave a locally hardened or heat affected objectionable zone on the fitting, unless this zone is carefully stress relieved. Thus valid objections stand against any of the working methods which are considered and it is apparent that more field experience with markings is needed to determine the best practical methods.

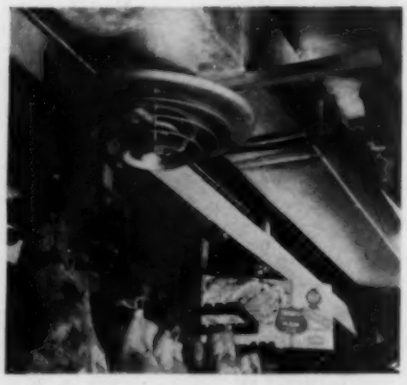
[This is the second of two articles by Mr. Kanter on this new ASTM specification. The first appeared in the April HPAC.]

MEAT PACKER SOLVES COOLING PROBLEMS

Use of high velocity air diffusers in a series of refrigerated rooms has solved certain problems of dehydration and has materially reduced cooling time in the plant of Stahl-Meyer, Inc., New York City, meat packers, by eliminating uneven temperatures.

In the sausage chilling room where pork sausage is set after the links have been stuffed, a temperature of 34 F is maintained. Under the previous system the setting of the product required from 12 to 24 hr, but with the new system in operation this time has been reduced to approximately 2 hr, it is stated.

According to tests made at the plant, dehydration of products, such as boiled ham and link sausage, has been reduced by 2 per cent. It was also found that the sausage retains its bloom for a longer period of time when on sale in a retail store. The product cools quickly, evenly and thoroughly, the appearance of the product is preserved, and its keeping qualities are enhanced. Temperatures are closely equalized throughout the boiled ham cooler, in spite of the close spacing of the meat in the racks, it is reported.



—Molitor photo, courtesy Anemostat Corp. of America.

Uneven temperatures eliminated

Another important point in connection with the installation is that formerly employees working in the low temperature rooms were uncomfortable because of excessive drafts. Since air velocities have been equalized throughout the room, employees are now able to work in comparative comfort, even at low temperatures, it is said.

THE WATER-AIR CHART

Its Use for Solving Problems Involving the Exchange of Heat Between Air and Water ..... By William Goodman*

THE water-air chart shown on p. 222 of the April HPAC provides a graphical method for solving a great many types of problems dealing with the exchange of heat between air and water. The chart itself and its use were explained last month, and a table showing the values of N and θ was presented. This month, the solutions of various types of problems are explained with illustrative examples.

Heating in Parallel Flow

In most **air washers** the flow of air and the flow of water are parallel to each other. Example 1 illustrates the solution of a problem involving the heating of air in parallel flow with water.

Example 1—Air at an initial wet bulb temperature of 50 deg is to be heated in a parallel flow spray chamber with water at an initial temperature of 115 deg. 0.33 lb of water per lb of air† is to be used. The barometric pressure is 28 in. Find: (a) the highest final wet bulb temperature to which the air can theoretically be heated; (b) the final wet bulb temperature of the air and the final temperature of the water if the efficiency of the apparatus is 75 per cent.

Solution—

a) The skeleton chart of Fig. 4 (p. 224, April HPAC) illustrates the method of using the water-air chart of Fig. 1 (p. 222, April HPAC) to solve this problem. Locate point *1* where the horizontal enthalpy line corresponding to the initial wet bulb temperature of 50 deg intersects the vertical line representing the initial water temperature of 115 deg.

From equation 4 on p. 224, April HPAC, $N = 0.33$. Using Table 2 (p. 223, April HPAC) for $N = 0.33$, $\theta = 28°\text{-}6'$. Draw line *1-3* through point *1* at an angle of 28°-6′ with the horizontal. The highest final wet bulb temperature to which the air can theoretically be heated is read at point *3* where line *1-3* intersects the saturation curve for a barometric pressure of 28 in. The highest possible final wet bulb temperature of the air is 70.3 deg as read at point *3*. This is also the lowest final temperature to which the water can theoretically be cooled.

b) Locate point *2* on line *1-3* in such a manner that the length of line *1-2* is 75 per cent of the length of line *1-3*. The final wet bulb temperature of the air will be 66 deg as read where the horizontal enthalpy line through point *2* intersects the saturation curve for 28 in. The final temperature of the water will be 81.7 deg as read on the vertical line through point *2*.

Cooling in Parallel Flow

The following example illustrates the method of solving a parallel flow problem in which air is cooled by water.

Example 2—Air at an initial wet bulb temperature of 78 deg is to be cooled in a parallel flow process by water at an initial temperature of 52 deg. The barometric pressure is 29 in. The water used amounts to 0.55 lb per lb of air. Find: (a) the lowest final wet bulb temperature to which the air can theoretically be cooled; (b) the final wet bulb temperature of the air and the final temperature of the water if the efficiency of the apparatus is 85 per cent.

Solution—

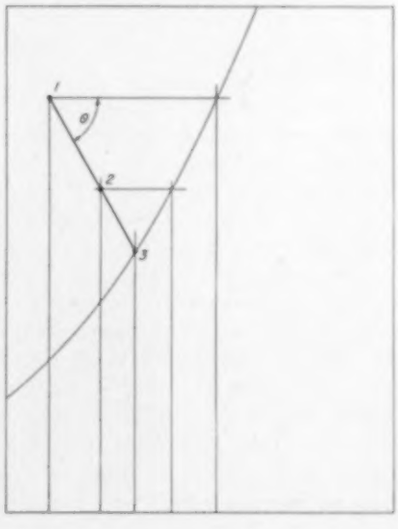
a) Fig. 6 illustrates the use of the water-air chart for this example. Locate point *1* where the horizontal line representing the initial enthalpy of the air—corresponding to the initial wet bulb temperature of 78 deg—intersects the vertical line representing the initial water temperature of 52 deg.

From equation 4 on p. 224 of the April HPAC, $N = 0.55$. Referring to Table 2 (p. 223, April HPAC), for $N = 0.55$, $\theta = 41°\text{-}40'$. Through point *2* (see Fig. 6) draw N-line *1-2* at an angle of 41°-40′ with the horizontal. This N-line intersects the saturation curve corresponding to the barometric pressure of 29 in. at point *3*.

The lowest final temperature to which the air can theoretically be cooled is 68.4 deg as read at point *3*. This is also the highest final temperature to which the water can theoretically be heated.

b) Locate point *2* on line *1-3* of Fig. 6 in such manner that the length of line *1-2* is 85 per cent of the length of line *1-3*. The final wet bulb temperature of the air will be 70 deg as read where the horizontal enthalpy line through point *2* intersects the saturation curve for 29 in. The final water temperature will be 66 deg as read at point *2*.

Fig. 6—*N*-line *1-3* represents the cooling of air by cold water. Angle θ is measured from the horizontal



Symbols

c' = specific heat of liquid.
G' = weight of liquid circulated, lb per lb of dry air.
h = enthalpy of saturated air at wet bulb temperature t'.
N = see Table 1 on p. 221, April HPAC, for equations for N.
t_1 = temperature of liquid, F.
t' = wet bulb temperature of air, F.
θ = angle for N-line.

Numerical subscripts refer to condition of air or liquid at various points in heat transfer process. For example, subscripts 1 and 2 refer respectively to initial and final conditions of air or liquid.

*Consulting engineer, The Trane Co. Member of HPAC's board of consulting and contributing editors.

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†The following relationship between *pounds of water per pound of air* and *gpm per 1000 cfm* will frequently prove useful:

$$\text{gpm}/1000 \text{ cfm} = 9\ G'$$

where G' is weight of water circulated, pounds per pound of dry air. This relationship is true only for standard air having a density of 0.075 pounds per cubic foot.

Heating in Counterflow

In vertical cooling towers, counterflow of air and water is widely used. The water-air chart will prove particularly illuminating in the solution of counterflow problems and in showing the limiting conditions that are obtainable. In counterflow, the

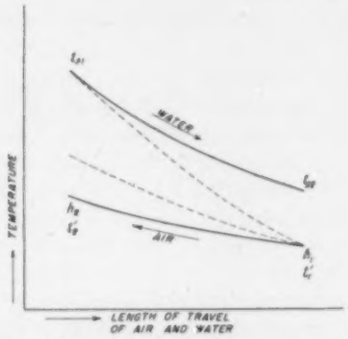


Fig. 7—Heating air with warm water in counterflow. The dotted lines represent one of the limiting cases. For the limiting case illustrated here, the water is cooled to the initial wet bulb temperature of the air

limiting conditions are difficult to visualize without the chart.

In parallel flow the initial temperatures of the air and water intersect at a point on the water-air chart. In counterflow—on the other hand—this is not the case. As can be seen in Fig. 7, the air at the *initial* wet bulb temperature is in contact with water at the *final* temperature. Unless the final temperature of the water is known, the point representing the intersection of the initial enthalpy of the air with the final temperature of the water cannot be drawn on the water-air chart. All that can be done is to first draw on Fig. 8 line *3-6* representing the initial enthalpy of the air, and line *4-7* representing the initial temperature of the water. These two lines form a right angle. Now, before the *N*-line can be drawn, one of the following quantities must be known: (a) the final temperature of the water; (b) the final wet bulb temperature of the air; or (c) the efficiency of the process. Thus, in Fig. 8 if the final water temperature is known, it is evident that point *1*, representing the intersection of the final water temperature with the initial enthalpy of the air, can be located. If the *N*-line is now drawn through point *1*, the final enthalpy of the air can be read at point *2* where the *N*-line intersects the vertical line representing the initial water temperature.

If the final water temperature were represented by point *3* instead of point *2*, a second *N*-line parallel to the first one could be drawn as in Fig. 8. In this case, the final enthalpy of the air would be represented by point *4*. Thus, it is apparent that, unlike parallel flow, a different *N*-line must be drawn for each final temperature of the water or final enthalpy of the air. In parallel flow, all of the points representing the final conditions lie on the same *N*-line. In counterflow, a different *N*-line must be drawn although all of the *N*-lines will be parallel to each other.

If the exchange of heat between the air and water is represented by a line such as *1-2* of Fig. 8, then the intermediate temperatures of the air and water can be read at a point on the same line *1-2*. Thus, if at some point in the process, the water temperature is represented by the vertical line through point *5*, the enthalpy of the air can be read on the horizontal line passing through the same point.

The limiting condition (that is, the highest final wet bulb to which the air can theoretically be heated in a perfect apparatus) is obtained when the water is cooled to as low a temperature as is possible. It is apparent that for the case illustrated in Figs. *7* and 8, the lowest possible temperature to which the water can theoretically be cooled is the initial wet bulb temperature of the air. Consequently, point *6* in Fig. 8 is located where the vertical line of water temperature which is equal to

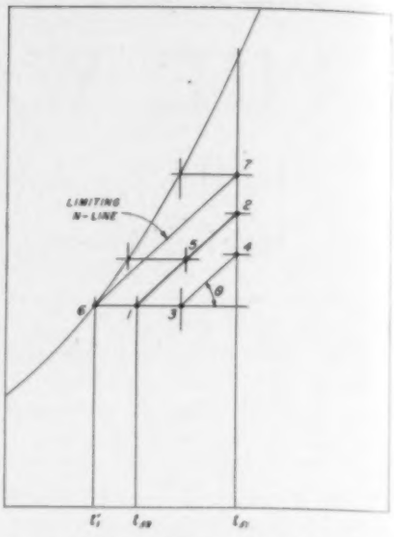


Fig. 8—*N*-line *1-2* represents the heating of air by warm water in counterflow. The highest theoretical wet bulb temperature to which the air can be heated is read at point *7* on the limiting *N*-line *6-7*

the initial wet bulb temperature crosses the saturation curve. This point represents the intersection of the final water temperature with the equal initial wet bulb temperature. By drawing an *N*-line through point *6* parallel to the other *N*-lines, the highest final wet bulb temperature to which the air can theoretically be heated may be read at point *7*, which lies at the intersection of the limiting *N*-line with the vertical line of initial water temperature.

The *N*-line *6-7* illustrated in Fig. 8 represents only one type of limiting *N*-line. There are other types which may be drawn from different points on the saturation curve. These are discussed in more detail later.

The efficiency of a counterflow process is represented by the ratio of the length of the *N*-line for the actual process to the *N*-line for the limiting case. Thus, in Fig. 8, for the process represented by line *1-2*, the efficiency would be equal to the ratio of the length of line *1-2* to line *6-7*.

Example 3 illustrates the solution of a problem in counterflow.

Example 3—Air enters the bottom of a vertical cooling tower at an initial wet bulb temperature of 75 deg. The water entering at the top of the tower is 93 deg. The weight of water circulated is 0.5 lb per lb of dry air. The barometric pressure is 27 in. Find: (a) the final wet bulb temperature to which the air will be heated if the water is cooled to a final temperature of 79 deg; (b) the

SUMMARY—The water-air chart provides a graphical method by which a great many types of problems dealing with the exchange of heat between air and water can be greatly clarified and conveniently solved for numerical answers. It can be used not only for problems involving the exchange of heat between air and water in spray chambers and coils, but also for problems dealing with the temperature of the spray water in evaporative condensers and spray coolers, and for problems dealing with the surface temperatures of coils. . . . . The water-air chart provides a particularly clear picture of the action taking place in air washers, cooling towers and coils. With its aid many difficult problems can readily be visualized and solved. This is particularly true of counterflow problems, otherwise difficult to visualize

temperature of the water in the tower at the point where the wet bulb temperature of the air is 78 deg; (c) the highest final wet bulb temperature to which the air can theoretically be heated; (d) the efficiency of the process in part (a) of this example.

Solution—

a) Fig. 8 illustrates the use of the water-air chart (shown on p. 222, April HPAC) for this problem. Locate point *6* at the intersection of the vertical line of 75 deg with the saturation curve for a barometric pressure of 27 in. Through point *6* draw the horizontal line *6-3* representing the initial enthalpy of the air. Draw the vertical line *4-7* representing the initial water temperature of 93 deg. Locate point *1* where the vertical line representing the final water temperature of 79 deg crosses the horizontal line *6-3*.

From equation 4 on p. 224 of the April HPAC, $N = 0.5$. From Table 2 (p. 223, April HPAC) for $N = 0.5$, $\theta = 38°\text{-}58'$. Draw the *N*-line through point *1* at an angle of 38°-58' with the horizontal. The final enthalpy of the air may be read at point *2* where the *N*-line intersects the vertical line of initial water temperature. The final wet bulb temperature of the air is read where the horizontal line of final enthalpy through point *2* intersects the saturation curve for 27 in. The final wet bulb temperature of the air will be 81.2 deg.

b) Locate the point where the vertical line of 78 deg wet bulb intersects the saturation curve for 27 in. barometer. Extend a horizontal line through this point until it intersects *N*-line *1-2* at point *5*. The water temperature as read at point *5* will be 85.5 deg.

c) To determine the highest final wet bulb temperature to which the air can be heated, draw through point *6* an *N*-line which is parallel to line *1-2*. This second *N*-line intersects the vertical line *4-7* of initial water temperature at point *7*. The horizontal line through point *7* represents the highest possible final enthalpy of the air. The highest final wet bulb temperature of the air is read where the horizontal enthalpy line through point *7* intersects the saturation curve for 27 in. The highest possible final wet bulb temperature to which the air can theoretically be heated is 82.8 deg.

d) The efficiency of the process as determined from the ratio of the length of line *1-2* to line *6-7* is 78 per cent.

Limiting *N*-Lines in Counterflow Heating

The case of the limiting *N*-line for which the final water temperature is cooled to the initial wet bulb temperature of the air has been discussed in the preceding section. There are other cases in which the final wet bulb temperature of the air may theoretically be equal to the initial temperature of the water. There are still other cases in which the wet bulb temperature of the air and the temperature of the water may theoretically be the same at some intermediate point in the process. In other words, the initial and final temperatures of the air and water differ, but they are the same at some intermediate point. These cases, which are illustrated in Figs. 9 and 10, all depend upon the slope of the *N*-lines, which in turn depend upon the weight of water.

On the saturation curve the wet bulb temperature and the temperature of the water are always equal. For any point to the right of the saturation curve, the water temperature is higher than the wet bulb temperature of the air. Obviously, air can be heated by water only as long as the temperature of the water is higher than the wet bulb temperature of the air. The transfer of heat ceases whenever these two temperatures become equal. Consequently the limiting *N*-line is obtained whenever the first *N*-line, which lies wholly to the right of the saturation curve, just touches the saturation curve—regardless of whether the *N*-line passes through one of the terminal temperatures or through an intermediate temperature.

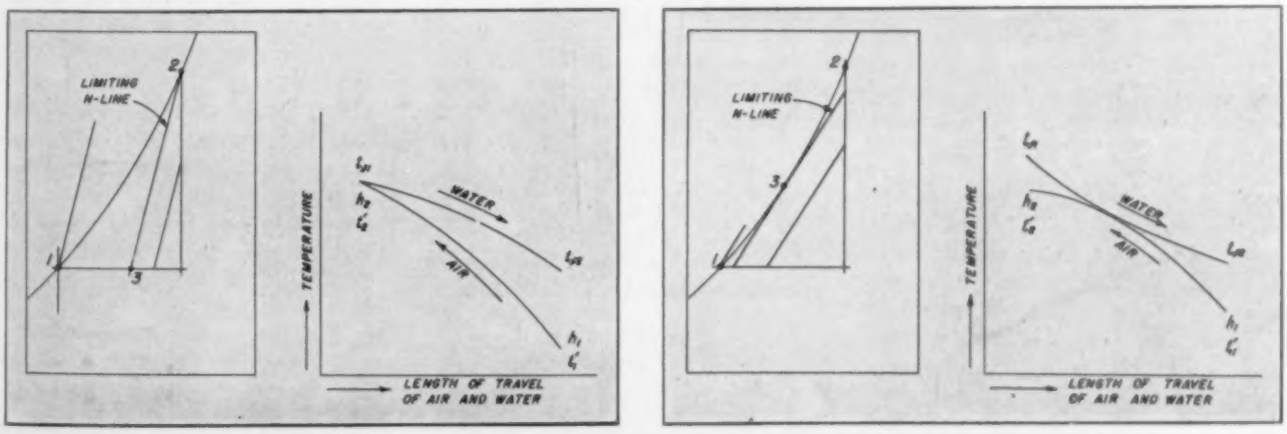
Although limiting *N*-lines can be drawn through any point on the saturation curve which lies between points *1* and *2* of Figs. 9 and 10, it is obvious that of all the parallel *N*-lines that can be drawn, only one will lie wholly to the right of the saturation curve. It is important to realize that the limiting line must lie wholly to the right of the saturation curve; if any portion of it lies to the left of the curve, that portion of the limiting line will obviously represent an impossible condition for heating—one where the temperature of the water is lower than the wet bulb temperature of the air. Of a series of parallel *N*-lines, the one *N*-line which touches the saturation curve without any portion of it extending to the left of the saturation curve is the limiting *N*-line from which the theoretical final temperatures should be computed.

Cooling in Counterflow

In counterflow cooling, the *N*-lines all lie to the left of the saturation curve. All of the discussion in regard to the various uses of

Fig. 9—Location of limiting *N*-line for case in which the final wet bulb temperature of the air reaches the initial temperature of the warm water in counterflow heating. The lowest theoretical temperature to which the water can be cooled is read at point *3* on limiting *N*-line *2-3*

Fig. 10—Location of limiting *N*-line for case in which the wet bulb temperature of the air becomes equal to the temperature of the water at some intermediate point in counterflow heating



N-lines for counterflow heating also apply to counterflow cooling.

In counterflow cooling the limiting N-line of Fig. 11 must pass through either point *1* or *2* which represents the combination of terminal temperatures. Unlike the limiting N-lines for heating air, a limiting N-line for

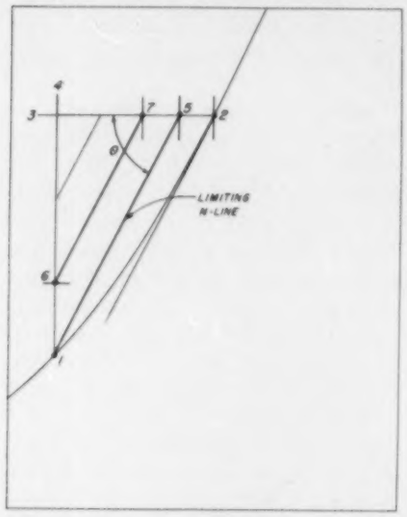


Fig. 11—Location of limiting N-line for case in which the final wet bulb temperature of the air becomes equal to the initial temperature of the cold water in counterflow cooling. The highest temperature to which the water can be heated is read at point *5*

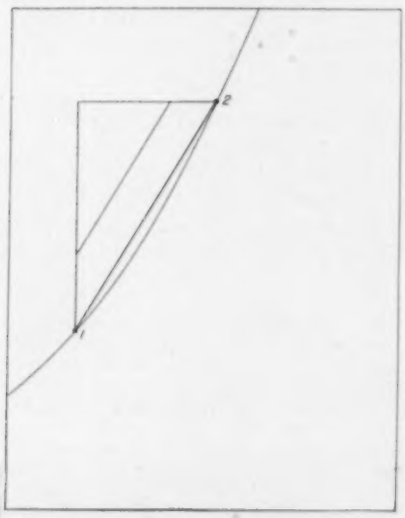


Fig. 12—Limiting N-line for counterflow cooling in which initial and final temperatures of air and water are equal

cooling air cannot touch any point on the saturation curve between the terminal points *1* and *2* without extending to the right of the saturation curve. Of a series of parallel N-lines, the one N-line which passes through one of the terminal points and which lies wholly to the left of the saturation curve is the limiting N-line from which the theoretical final temperature may be found for counterflow cooling.

Although the limiting N-line cannot touch any point on the saturation curve between the terminal points *1* and *2* of Fig. 11, it is obvious from Fig. 12 that an N-line can pass through both points *1* and *2* without extending to the right of the saturation curve. Consequently for cooling air, the limiting N-line must pass through one of the terminal points or through both of the terminal points, but it cannot pass through any point on the saturation curve between the terminal points without extending to the right of the saturation curve.

The solution of a problem in counterflow cooling is illustrated by example 4.

Example 4—Air at an initial wet bulb temperature of 72 deg is to be cooled by water at an initial temperature of 48 deg. The weight of water supplied is 0.8 lb per lb of dry air and the barometric pressure is 28 in. Find: (a) the lowest theoretical final wet bulb temperature of the air and the highest theoretical final temperature of the water. (b) the final wet bulb temperature and the final water temperature if the efficiency is 80 per cent.

Solution—

a) Referring to Fig. 11, horizontal line *2-3* corresponds to the initial enthalpy of the air at a wet bulb temperature of 72 deg and a barometric pressure of 28 in. The vertical line *1-4* represents the initial water temperature of 48 deg.

From equation 4 on p. 224 of the April HPAC, $N = 0.8$. From Table 2 on p. 223 of the April HPAC for $N = 0.8$, $\theta = 52°\text{-}18'$. Obviously, for this N-line, the air can theoretically be cooled to the initial water temperature of 48 deg; that is, to point *1* of Fig. 11. The limiting N-line through point *1* is line *1-5*. The highest theoretical possible final water temperature is 69.7 deg as read at point *5*.

b) Draw line *6-7* parallel to *1-5*. So locate line *6-7* that its length will be 80 per cent of the length of line *1-5*. The final wet bulb temperature of the air as read at the point where the horizontal enthalpy line through point *6* intersects the saturation curve for 28 in. is 53.9 deg. The final water temperature as read on the vertical line through point *7* is 65.4 deg.

Visualizing Location of N-Lines

As an aid to memorizing the location of N-lines for the problems just described, refer to Fig. 13. Notice that the initial temperatures for

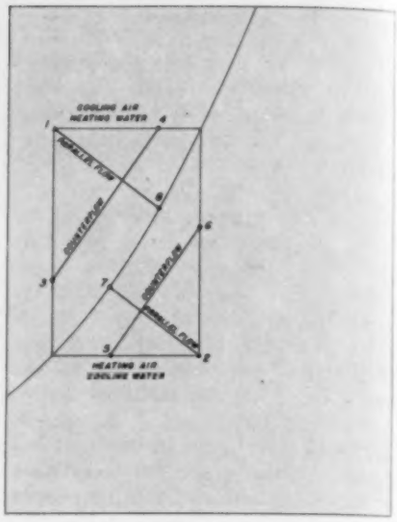


Fig. 13—Rectangle for visualizing locations of N-lines

heating and cooling form a rectangle. Parallel flow cooling or heating always takes place along N-lines drawn only from points *1* and *2*. Counterflow cooling or heating takes place along diagonals such as *3-4* or *5-6*.

[*This is the second of a series of articles on the water-air chart.*]

RADIANT HEATING

THE EDITOR—

F. W. Hutchinson's article on radiant heating in the April HPAC was indeed most interesting. I for one am particularly glad he has pointed out that there are factors other than the radiant comfort theory which greatly affect the selection and design of such heat exchange surface. For example, he pointed out that other considerations such as the relative responsiveness to control and flexibility of operation and the ability of the panel fully to utilize the circulating fluid in the energy exchange, as well as the other factors of maintenance, repair and costs, are also important.

However, I personally feel that responsiveness to control and flexibility of operation are really paramount points. Using, of course, the cast iron type of panel he described you do get greater flexibility of control in more rapid cooling, and since this is possible there is far less residual heat in the panels, which I think is an important matter.—CARL F. BOESTER, director of housing research, Purdue University.

Proper Operation at Minimum Cost If Piping Layout Is the Right One

The Chief Has Gard Analyze the Piping for Their New Diesel Engine Cooling System . . . By A. C. Kirkwood

PIPING LAYOUT — In past months, you've listened in on other piping discussions of the Chief and Gard, as reported by A. C. Kirkwood of the Burns & McDonnell Engineering Co. (consulting engineers) and a member of HPAC's board of consulting and contributing editors. This month, they take up the question of the piping design and layout for the new diesel engine cooling system at their plant. It comprises a two circuit installation, as there is an inner or jacket water system and an outer or raw water system. The jacket water is cooled in coils by means of the raw water, and the raw water is cooled in towers. . . . For economical operation under the various conditions of load and wet bulb temperatures that must be met, it is essential that the piping be sized properly to enable the best use of the investment in pumps. This is done by Gard in the report he prepares for the Chief by plotting the curves shown in Figs. 3 and 4. How curves of this kind are used in such problems is explained in the text

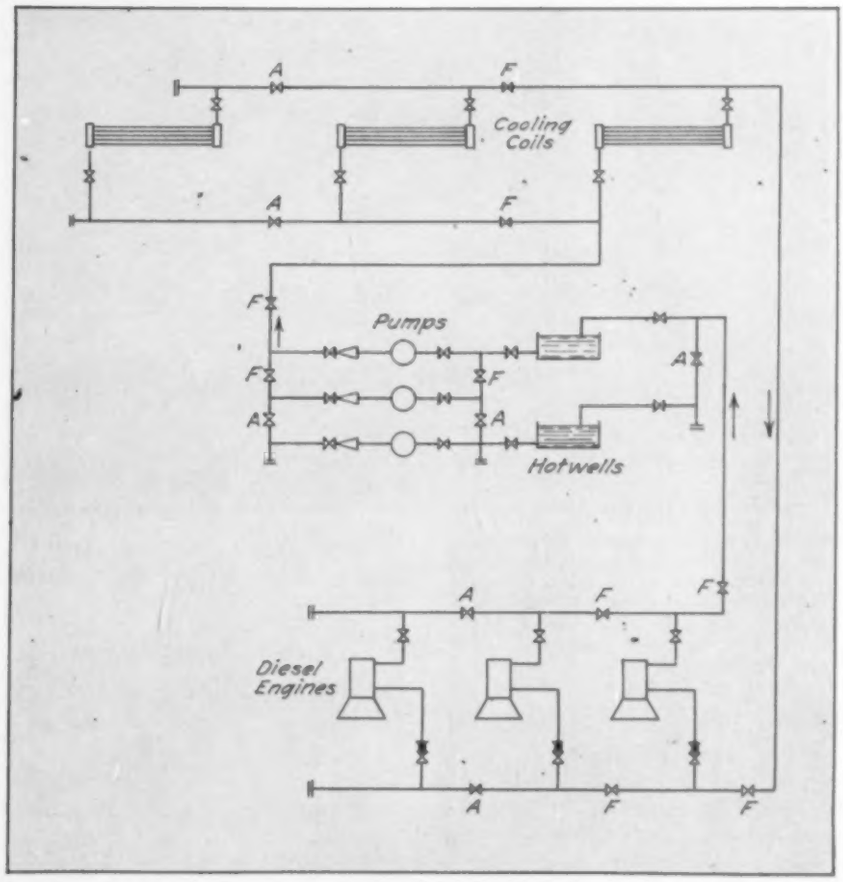


Fig. 1—The inner or jacket water piping circuit. Hot water from the engines goes to the hotwells, is pumped through the cooling coils, and from there goes back to the engines. Valves marked *A* serve as isolators for use in making further extensions and thereafter will serve as header sectionalizing valves. Valves marked *F* will be required in the future

CHIEF, I wonder if the consulting engineers designing our new diesel plant cooling system have the correct idea on the piping layout?"

"Well, what's the trouble, Gard?"

"Oh, they pipe it up as though the cooling system was complicated. Frankly, I can't see anything but a few pumps and cooling coils and engines. Then, too, they have larger piping than I think is necessary. I wish you'd look over these plans, Chief."

"I already did, Gard. Before you criticize too severely, perhaps you should examine the reasons for their design. These specifications and prints for the cooling towers, engines and pumps have just arrived. Read them over, then analyze the operating conditions in the light of these wet bulb variation tables and this expected load analysis. Write me a short report on your findings, and then let's see what your criticisms amount to."

* * *

Cooling tower and engine jacket water piping must meet several distinct sets of operating conditions. To accomplish this successfully, the designer must analyze correctly the different sets of conditions and the operator must follow the designer's intended operating procedures. The penalty for failure on the part of either is adverse water temperatures and probable loss of money in investment and operating costs.

Assuming that the correct engines have been selected, and that the cooling tower has been selected to meet varying engine requirements, the problem confronting the piping designer is one of adequacy, reliability and flexibility.

The first step is acknowledgment of the specific major requirements. The mere fact that a cooling tower is used indicates a lack of other source of adequate cooling water. It admits dependence upon atmospheric wet bulb temperatures. This

temperature varies with the seasons, and consequently operation of the tower must be varied to conform. In most cases the operating load will vary during the day and the year. Flexibility of tower operation must be provided for by proper piping arrangements.

Although pressures and temperatures are low, the piping is subject to the usual elements of deterioration and injury; minimum safe-

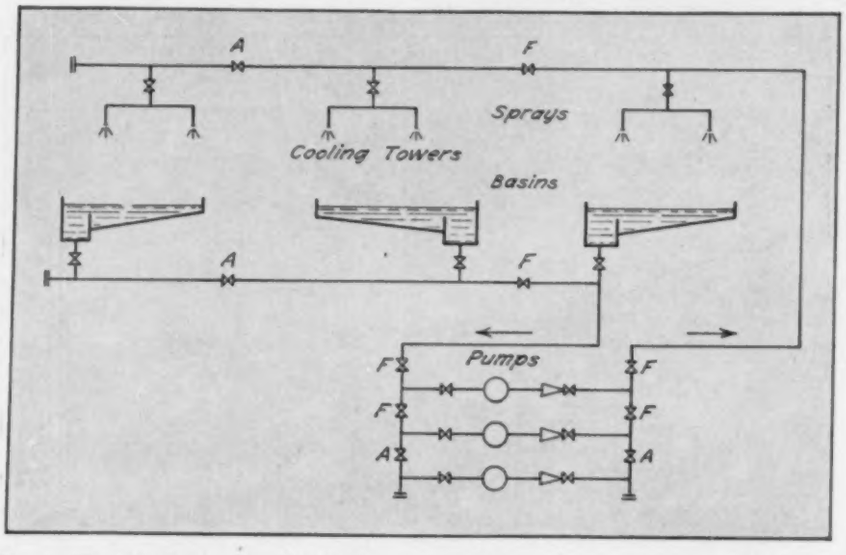


Fig. 2—The outer or raw water piping circuit. The source of heat is the cooling coils of the jacket water system shown in Fig. 1. The hot raw water is collected in the basin and sump at the bottom of the tower. The valve designations *A* and *F* have the same meaning as in Fig. 1

guards to insure uninterrupted service are therefore required. Needless to say, piping of adequate size must be installed. If an early increase in capacity is expected the original piping should permit future additions without wastage of the original investment. A word of caution: Requirements of flexibility of engine, tower and pump operation are essential; elimination of false or unwarranted friction heads is necessary. Piping friction having a low order of variation is a desirable aid to proper operation.

Other factors may influence piping design and operation. For instance, fuel oil storage tanks can be (and frequently are) heated by engine jacket water during months of low temperature. Some building or space heating may be done by the jacket water.

Several types of systems employing cooling towers are in use. General classification divides them into two groups—single and double circuit—with the double circuit group being the more desirable under most circumstances. In this group the design employs an inner or jacket water circuit so that the same water is always recirculated through the jackets, thus eliminating possibility of scale deposits. The outer circuit recirculates raw or (sometimes) treated water to which is added necessary make up for use in the cooling tower proper. Piping is thus required for two distinct services.

Two Circuit System

Schematic diagrams of a two circuit cooling system are shown in Figs. 1 and 2, the former being the inner or jacket water system and the other the outer or raw water system.

Considering first the jacket water system, it is seen that the source of heat is three engines and the heat dissipation units are cooling coils. Circulation is by means of three jacket water pumps and for purposes of this discussion they are assumed to be identical. Hot water from all engines is discharged to either or both hotwells at the will of the operators, whence it enters the suction header. Any one or more of the jacket water pumps passes the water through the cooling coils and on to the appropriate engines. The installation is arranged for future additions, and consequently provisions are included to permit connections to future equipment without undue shutdown of existing equipment or loss of service. Since all services are well below rating of valves and because only a few operations a day would be the maximum expected, the full reliability and flexibility gained by installing a loop header system can be deferred until the future, when additional major equipment is to be installed.

Valves marked *A* serve as isolators for use in making future extensions and thereafter will serve as header sectionalizing valves. Those marked *F* will be required in the future as sectionalizing valves but may be omitted for the present. If omitted in the original installation it is possible to install spool pieces which have the same face-to-face as the valves, thereby reducing outage time and cost of future installation.

Before considering the actual conditions involved in a typical jacket water problem, it is well to consider the basic requirements of the raw water circuit. Fig. 2 shows a typical layout. The source of heat is the cooling coils of the jacket water system. The hot raw water is collected in the basin and sump at the bottom of the tower. After passing through screens in the sump or pump suction chamber to remove leaves, papers and other trash the water enters the raw water pump suction header. Raw water pumps discharge at sufficient head to eject the water through spray nozzles or place it in the spray basin or troughs at the top of the tower. From here the water passes through the air flow created by the fans and is cooled before the falling water reaches the outer surfaces of the cooling coils. Upon reaching the coils, the water is again heated and, as hotter water, is again collected in the basin to repeat the cycle

As in the case of the jacket water circuit, there are multiple elements to be considered. They are the tower units or cells and the pumps. The piping shown is intended to provide adequate yet simple connections for this layout, with proper provisions for future expansion. Valve markings are the same as for the jacket water circuit of Fig. 1.

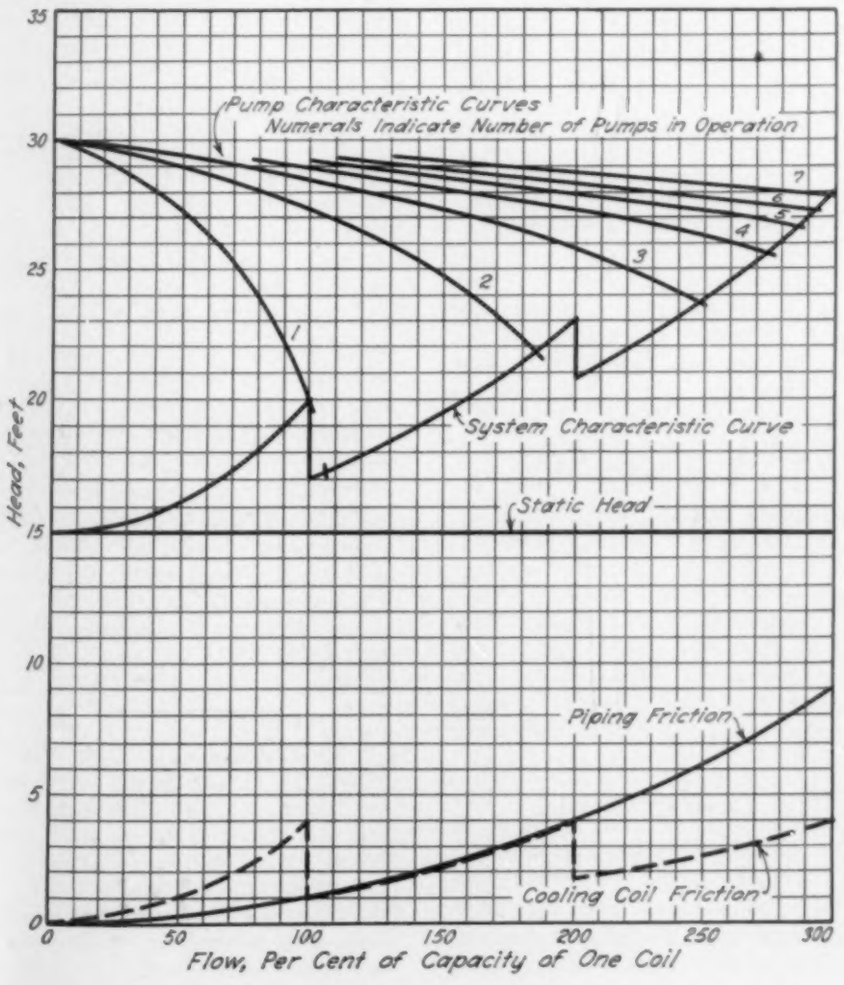
Why Piping Is Important

The degree of satisfaction achieved in operating the cooling system depends materially upon the ease with which the piping layout

responds to the varying operating conditions. This response is measured not only by the flexibility and reliability of the layout, but also upon the proper selection of pipe sizes. Pumps will not adequately serve multiple unit cooling towers if pipe sizes are too small unless either a large number of small pumps or a few large pumps are installed. The ideal situation would be where the operation of another pump would result in a proportionate increase in flow. The characteristics of flow in piping and economic considerations prevent complete attainment of this goal, but a desirable approach to perfection can be made.

In Fig. 3 two sets of curves are shown—the system and pump characteristic curves. The former is made up of static head, piping friction and coil friction, and the characteristics of each are indicated separately. Static head is constant. Coil friction, identical for each of the three coils indicated, varies with coil flow. On the other hand, piping friction varies with the total system flow. Static head and coil friction are more or less set by design conditions other than those under control of the piping designer. However, selection of piping may (and usually does) determine the pumping conditions and the definite operating characteristics of the entire system.

Fig. 3 shows that if a three element tower with jacket water piping similar to that diagrammed in Fig. 1 were installed, the shut off head would be 30 ft to accommodate the 28 ft head when three coil units were in service. This is based on one of the identical pumps serving one coil, the pumps having a shut off head of 150 per cent rated head. Such a pump would have what is considered a steep "head-capacity characteristic." Further reference to the curve shows that two pumps would serve about 185 per cent the capacity of one coil (92.5 per cent of two coils) and that three pumps would serve about 246 per cent the capacity of one coil (or 82 per cent of each coil).



Similarly, seven pumps would be required to serve three coils. This condition is not desirable.

Decreased Pipe Friction

The results of decreased pipe friction are shown in Fig. 4. Increasing pipe sizes has reduced the piping friction 50 per cent, with the static head and the coil friction remaining the same. In this case the pump shut off head is but 30 per cent above rated head and the pump characteristic is more desirably flat. As would be expected, the required number of pumps is reduced. Two pumps will deliver water for approximately 95 per cent of the capacity of two coils and three pumps will handle about 258 per cent of one coil. However, only five instead of seven pumps are necessary for three coils—a saving of nearly 30 per cent in pump investment and a material saving in wiring and piping connections for the pumping units. Building space is likewise conserved. These reductions are directly attributable to more desirable pipe sizes.

Should the pipe sizes used in Fig. 4 be combined with the pumps used in Fig. 3, the number of pumps required for three cell operation would be reduced to four.

If the ring header should be completed the piping friction would be reduced further. In fact, it would decrease to a point where two pumps would supply virtually the full flow to two coils, and three pumps would very closely approach the flow for which each of the three coils was designed.

Vagaries of wet bulb temperature and operating load are such that the operator must be alert constantly to the results of changing conditions. Wet bulb temperatures varying from low values (the minimum depending upon locality of the installation) to a maximum of 80 F or more are encountered. Under these conditions he must maintain uniform jacket water temperatures even though the

Fig. 3—System and pump characteristic curves. The system characteristic curve is the total of the cooling coil friction and the piping friction (shown at the bottom) and the static head of 15 ft. The left third of the graph is for one cooling coil in use, the middle third for two cooling coils in use and the right third of the graph for three cooling coils in use. Note that seven pumps would be needed to serve all coils at full capacity

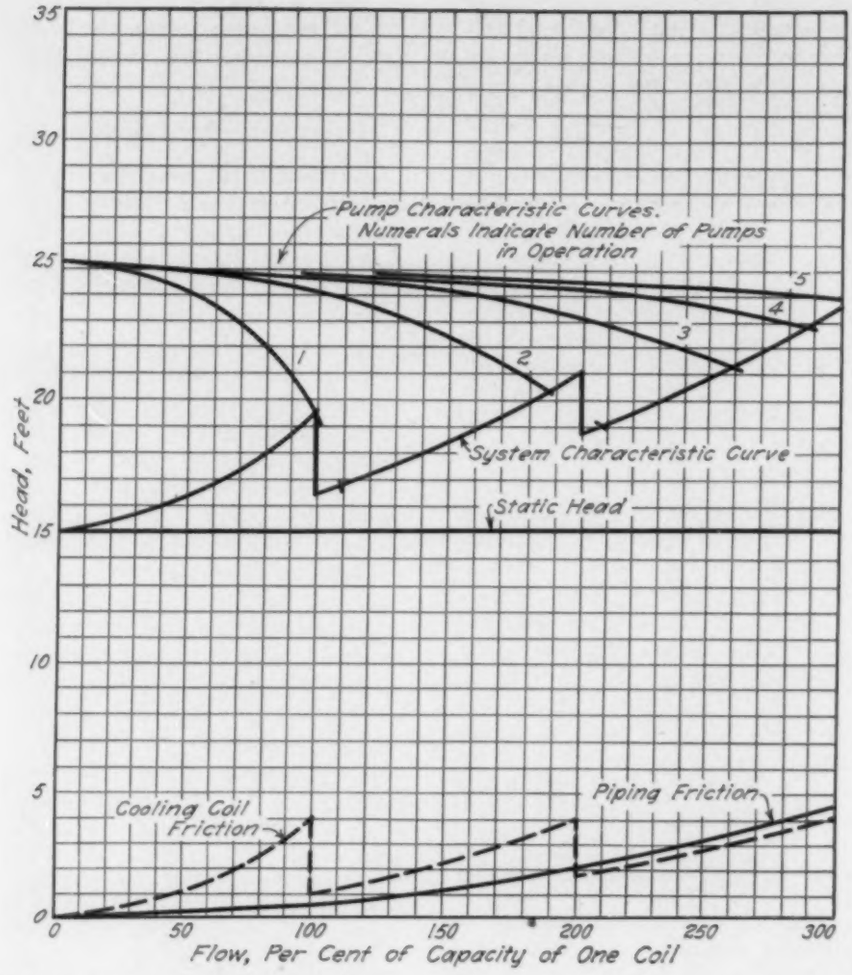


Fig. 4—This is the same sort of a graph as Fig. 3, but the piping sizes have been increased so that the piping friction is 50 per cent of what it is in Fig. 3. Note that now only five pumps—instead of seven pumps — are required for full capacity operation of the three coils

load is varying. The operator and the designer necessarily recognize the importance of piping friction of a low order of variation.

Other Piping Details

In addition to the connections indicated in and inferable from Fig. 1, the designer may install heating coils in fuel oil tanks or in buildings. These units should be treated as additional cooling coils. Not shown, but often included, is a tempering bypass between hot discharge from the jacket water pumps to cold discharge from the coils. This bypass may be either hand or automatically controlled to regulate the temperature of the supply to the jackets.

Omitted from Fig. 2 is a raw water make up line to each basin. This may be float controlled, but frequently is hand controlled with water level gage lines brought into the plant. In the latter installation the make up valves are usually adjacent to the gage glasses. In some installations the sprays in each cell or unit may be in parallel with operation permitting either or both sets to be operated at one time. The only major valving required would be shut off gate valves.

Freezing presents a definite problem, since much of the piping is either exposed or above the frost line. Several solutions to this problem have been devised, one of the more desirable being the drainage of lines and equipment exposed to freezing temperatures. Where shut off valves are situated so that parts of the adjacent lines are subject to freezing, but are not in the drained section, it is desirable to provide weeps to maintain a small amount of circulation of warm water. Drainage emphasizes the necessity of venting. Vent and drain valves should be placed so that operators can manipulate them without becoming wet from sprays in the raw water circuit.

Cleaning Facilities

Cleaning facilities are essential. The jacket water system recirculates one charge of water with make up limited to a negligible amount. The hotwells should be provided with sludge pumps and drain valves. Engines are often provided with jacket drains to the hotwell and sometimes coils are drained to the hotwells. The latter conserves jacket water when the coils are drained to prevent freezing. Occasionally, jacket water hotwells are below sewer lines, in which case a small drain header serving all hotwells is connected to one of the pumps with a special branch discharge to the sewer.

Raw water basins and sumps require regular cleaning; the basins usually are sloped to a cleaning sump independent of the pump suction sumps. Drainage is to sewer by gravity or through one of the raw water pumps, by means of special connections, as relative elevations indicate as being necessary.

* * *

"There you are, Chief. Looks like I have gone off the deep end to support the engineers."

"Well, I wouldn't put it that way, Gard. But you have reached about the same conclusions that I had."

"You know, Chief, I'm still pinching myself over the fact that anything as simple as a cooling system like this would have to operate under so many different conditions, and require so many provisions to get foolproof operation the year around. Why, I didn't even include the list of various operating conditions and equipment combinations that might well be expected. I think the engineers have the right slant on this piping. Without much additional cost for piping we can hit the cooling water requirements right on the nose."

Of the 150,000 tons of refrigeration equipment in use in Texas, 66,000 tons is used for comfort cooling in 17 cities in the state of 25,000 population and over. These figures were given by L. P. Reiss in a talk on progress of refrigeration in Texas presented at the food preservation conference at the University of Texas.

"Open for Discussion"

Communications to the Editor

CONVERSATION PIECE

[Editor's Note—In his article entitled Air Change—Once a Major Criterion in the March HPAC, C. M. Ashley, director of research, Carrier Corp., came to the conclusion that the "air change" criterion has outlived its usefulness, that it is not a proper or accurate method of describing room air circulation, that its use may result in drafts, that it works in reverse as a criterion of ventilation, and that it is a poor choice even for crude estimating. At the end of his article appeared a short "filler" mentioning the Black Hole of Calcutta tragedy in 1756, when 123 of 146 English prisoners in a cell died because of lack of ventilation, excessive heat and humidity.

In a comment on p. 230 of the April HPAC, J. M. Frank, president, Ilg Electric Ventilating Co., stated that air change was certainly a "major criterion" in that situation of 1756, and it was not clear to him how any of the physical laws in existence at that time have been changed. Feeling it to be of interest and value to many readers, the ensuing comments of Messrs. Ashley and Frank are presented here.]

MR. ASHLEY—Concerning Mr. Frank's comment, it was not my purpose to intimate that there should be a lack of ventilation air or a lack of air movement, but that we should relate the ventilation requirement to the facts, rather than to an arbitrary standard which has no relation to the facts.

The case of the Black Hole of Calcutta is cited as a reason for using the "air change" criterion. I would like to suggest that it is rather an excellent example of where it does not work out. Had a designer of ventilation equipment for the Black Hole used the "air change" criterion, he would probably have selected something like a 15 min "air change," which would have resulted in about 0.8 cfm per person. While this is more than enough ventilation to reduce the carbon dioxide content to a livable level, it is far below the 10 to 30 cfm per person recognized as good practice from the standpoint of odor and smoke control. It is further entirely inadequate to remove the heat and moisture generated in this space without an excessive rise of temperature and moisture content.

Of course, if an air conditioning engineer had been called upon to provide equipment for the Black Hole with its occupancy of 146 men, he would not have designed it on an "air change" basis; he would have considered the amount of air necessary to maintain the desired conditions with regard to the number of people and other sources of load. In other words, he would have related it to the facts, rather than using an arbitrary criterion (air change) connected with the building construction.

MR. FRANK—I can see where this may lead to a long discussion—chances are, Mr. Ashley and I better meet at some point midway between Chicago and Syracuse and talk it out and save a lot of trouble.

My immediate reaction to Mr. Ashley's comment is that apparently we both look at problems in the light of our respective experiences, which is no doubt as it should be. I have been connected 31 years with a company that primarily sells ventilation and we have always stressed rapid air change, with of course some "relation to the facts," meaning type of occupancy, or type of process involved. Now, had one of our sales engineers been given the problem of ventilating a space the size of the Black Hole of Calcutta, with the requirement that it take care of 146 men, I feel sure that we would have tackled this as a rapid air change problem and while it might have been somewhat extreme, perhaps 10 air changes a minute, it would certainly have saved the lives of those men.

Mr. Ashley speaks of a 15 min air change. He will find if he refers to literature published by our company that in no case that we can think of do we recommend an air change as low as 15 min. Most of our published air changes for various types of places are from 2 to 5 min and even for residence attics, we recommend a 2 min air change on the entire upstairs cubical contents, because there, regardless of number of occupants, it is desirable to get both air change and air movement. In restaurants that have a high volume of cooking, we recommend as much as a ½ min air change and a great number of our most successful jobs are on that basis. It would be interesting to know how Mr. Ashley in the light of modern findings would have figured a system for the Black Hole of Calcutta on his basis.

MR. ASHLEY—I wonder if the difference of point of view expressed by Mr. Frank is not largely a matter of terminology. While Mr. Frank speaks of "air change", he clearly ties it up to occupancy, indicating that he is not talking about "air change" related to volume of room but rather to an entirely different factor; that is, cfm per person.

Of course, there is an intermediate position that tries to classify different types of jobs according to the general character of occupancy and then sets up an air change for each class of job. I am not altogether clear as to which method Mr. Frank is using. If he is relating his air handling directly to the number of people, then the use of the term "air change" is, I believe, confusing, unless he wants to term it "air change per person". I believe, however, that the term "cfm per person" is more accurately descriptive and avoids the danger of confusion which apparently has arisen in this case.

If the factor which Mr. Frank is using is related not only to the class of job but also to the cubic contents of the building, then I believe that the original comments which I made are still apropos. The best that can be said for this procedure is that it is a halfway approach to the factor which should govern the amount of outside air taken in and the amount of air movement in the space.

RADIANT HEATING WITH CAST IRON PANELS

THE EDITOR—

Since I have been associated with these cast iron panels for radiant heating from the time they were first designed by my friend A. H. Barker over 20 years ago, and since I have designed many other different types of metal panels used in connection with radiant heat installations, I was particularly interested in the article by F. W. Hutchinson in the April HPAC. Mr. Barker, after disposing of the patent rights for the embedded pipe method of panel heating, designed and patented the cast iron panels to provide a flexible method of radiant heat and a unit which could be applied to existing buildings without extensive structural alterations.

I think the first cast iron panels installed in America were erected in the library of the Pennsylvania College for Women, Pittsburgh, in 1932. These I ordered direct from England as they could not then be obtained in this country. They were installed to overcome complaints of drafts. The librarian found it impossible to keep warm and was continually catching colds, although the air temperature was always 72 F or above. After the panels were installed all complaints ceased, and during the eight years these have been in use, the results have been perfectly satisfactory.

Mr. Hutchinson implied in his article that the adoption of the cast iron panels or any other type of metal panel is not in accordance with the radiant heat comfort theory. This statement is somewhat misleading, it seems to me, because the whole theory of radiant heating is to take care of the heat losses from the human body; and while low temperature radiation appears to provide better comfort and a healthier condition than high temperature radiation, there is not as yet any very definite line of demarcation of surface temperatures.

The metal panels—whether made from cast iron sections, or fabricated from sheet steel plates to which are welded wrought iron pipes to form steam or waterways—are cheap to provide, easy to install and flexible in operation. Moreover, they do not present any great obstacle when changes in the internal construction of a building are necessary.

I installed a number of metal radiant heat panels in our work offices at Bethlehem, Pa., over five years ago, with perfect results. These panels are fabricated with steel plates which fit into some of the lower panels of the wood partitions. Coils of ¾ in. wrought iron pipe are spot welded to the steel plates and these are connected to the steam heating system in the usual manner. A radiant heat room thermostat controls the supply of steam according to the mean radiant temperature and the air temperature.

Radiant Heat for Powder Plants

At the present time I am designing a radiant heat system for several large powder plants for which sheet metal panels are contemplated. They are to be connected to one central steam supply.

The panels recommended are to be placed on the walls and will not occupy floor space which will be very valuable for the process work and, secondly, with radiant heat it is possible to provide comfort conditions with an air temperature 6 to 8 F below the air temperature necessary if an ordinary heating system is provided. This lower air temperature, coupled with the fact that the lower temperature means a higher relative humidity, should reduce the risk of fire.

The reason I recommended the radiant panels for the side wall was because, if the panels were placed in the floor and boxes of powder were placed on the floor, heat might generate below the boxes sufficient to be dangerous. The surface of a heated floor will soon become very hot if the radiation is stopped, as would be the case with boxes resting on the surface. I ruled out the question of radiant panels in the ceiling because these buildings are invariably constructed with a very light roof, which will be easily lifted in the case of an explosion. In the particular case in question, the roof will be high and not by any means airtight.

Mr. Hutchinson mentioned in his article that metal panels are regularly used for floor and wall as well as ceiling, but I have never seen these used for the floor except in systems known as "border" heating. The metal panels are then made narrow and used only in such places as aisles of churches or adjacent to walls and other places not usually occupied for standing or sitting. If the surface temperature is above 160 F, as Mr. Hutchinson mentions, it would be impossible to tolerate such high temperatures for the feet.

Radiant Heating Calculations

Concerning the question of calculations for radiant heating, it admittedly was customary in the early days of radiant heating to calculate the surface on the basis of heat transmission losses as in the case with convectional heating systems; but for many years heating engineers have based their calculations on the mean radiant temperature or MRT of the room compatible with the heat losses from the human body.

I have maintained for years that only when we realize that true comfort and health can only be maintained by taking care of the actual heat losses from the human body in a way that will meet the correct physiological and psychological reactions of the person shall we be on the right road to design correct heating and air conditioning systems.

It may be correct to say that in making calculations on the basis of heat losses through walls, etc., a close approximation may in some cases result; but on the other hand, it is a very dangerous procedure, because many disappointments will follow and the reputation of radiant heat will suffer. Those who have had no actual experience with radiant heat will be well advised to go carefully and if not sure should err on the right side, for once a radiant heat system is installed, it is a very

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expensive job to make additions or changes.

Capacity

Under the heading of "Capacity" Mr. Hutchinson remarked that "floor installations represent a compromise between radiant and convective heating." This is an argument of those using embedded pipes in the ceiling, but it is not substantiated by facts. The theory of radiant heat is to take care of the individual heat losses from the human body and the question as to whether the radiant panels are in the ceiling, floor or walls must be decided according to the type of building and the use to which the rooms are put.

The statement "Such advantages as accrue from radiant heating are therefore realized to a lesser extent as the panels are moved down from the ceiling", overlooks the fact that the floor surface must be much cooler than is permissible for either the walls or ceiling. It is true that a heated surface on the floor will give off more heat by convection than a similar surface on the ceiling, but this is more than counteracted by the lower floor temperature. This is substantiated by the results of my test taken at the Liverpool cathedral (as described in HPAC in July, 1938) where the air at high level was cooler than at low level.

In reference to the special panel in Fig. 6 of Mr. Hutchinson's article, it may be of interest for some to know that this type was originally designed several years ago for the large Shell Mex building overlooking the Thames in London. As additional heat was necessary to provide extra ventilation, this type of panel was introduced for the purpose of obtaining more convected heat for ventilation without raising the mean radiant temperature too high. —T. Napier Adlam, chief engineer, Sarco Co., Inc.

Reply by Mr. Hutchinson

The Editor—

Mr. Adlam states that my article "implied that the adoption of the cast iron panels or any other type of metal panel is not in accordance with the radiant heat comfort theory." In the opening paragraph of my article it was stated that the "selection of a particular type of panel is not dictated by radiant comfort theory . . .," while in the third paragraph appeared the statement, "If a cast iron system were operated at the surface temperature used with embedded panels the comfort characteristics, energy requirements and equilibrium performance of the two systems would be similar".

Commenting on the above "implication", Mr. Adlam discusses the comfort and health effects of high and low temperature radiation and states that there is not a definite "line of demarcation of surface temperatures". At no point in my article was the difference in comfort or health effects due to irradiation with energy from low or high temperature sources used as an argument either for or against the cast iron system.

Floor installation at Milan cathedral

In Mr. Adlam's comment there appears the statement that "if the surface temperature is above 160 F, as Mr. Hutchinson mentions . . ." My article did not state that the *surface* temperature is above 160 F.

As to the method of calculation and the data on heating effects given in my article, this information was obtained at the head office of a manufacturer of the cast iron panels; much of this information is available in engineering bulletins published in England and Italy by the manufacturers.

Mr. Adlam correctly states that "for many years heating engineers have based their calculations on the MRT of the room. . . ." This is the basis of the method described

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in my article and recommended by the two largest European manufacturers of cast iron panels. Transmission losses influence the inside equilibrium surface temperature and are therefore a factor in calculation of the MRT; the method here used involves the use of experimentally determined "equivalent panel ratings" which are non-rational coefficients that take into account the reduction of air temperature which is permissible with a higher MRT. This is brought out by the manufacturer's statement that if room air temperatures must be guaranteed the designer should use the actual heat dissipation rate of the panel rather than the "equivalent heating effect". The method described in my article is the one used in the great majority of English and continental cast iron panel installations.

Effect of Panel Location

The statement that wall or floor panels are less effective than ceiling panels for radiant heating is said by Mr. Adlam to be not substantiated by facts and an argument of those using embedded pipes in the ceiling. The most significant advantage claimed for radiant heating is that it permits realizing comfort in a room with reduced air temperature. Application of the heat transfer and heat balance equations from any text will provide unquestionable

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substantiation for the fact that the magnitude of the inside air temperature reduction decreases as the fraction of heat transfer by convection increases. Application of the same equations will also demonstrate that the advantage claimed for ceiling installation does not "overlook the fact that the floor surface must be much cooler than is permissible for either the walls or ceiling"; for the same MRT a lower temperature necessitates a larger area and the energy input to the low temperature floor system will exceed that to the warmer ceiling system. Practical substantiation for the claim of greater effectiveness of ceiling installations is available from the published ratings of cast iron panels. Reference to one manufacturer's catalog will show that the heating effect per Btu of actual energy input increases as the panel is moved from floor to wall to ceiling; obviously, therefore, it is the contention of the manufacturer that the effectiveness of the system, as measured by fuel consumption, will be greatest with a ceiling installation.—F. W. Hutchinson, instructor, department of mechanical engineering, University of California.

SPECIFICATION FOR WELDING FITTINGS

The Editor—

In his article on the new ASTM specification for welding fittings, the first part of which appeared in the April HPAC, J. J. Kanter has very ably clarified the reasons for the setting up of a new standard for welding fittings rather than to incorporate such requirements into existing standards such as covered by flange specifications. While the ASTM specifications are principally made to embrace materials in use and while welding flanges may be considered as welding fittings, yet a flange connection is entirely different from a purely welded assembly. Furthermore, the flange specifications embrace threaded flanges which are entirely outside the scope of welding structures. It is expected to include in these specifications materials used for crosses, socket weld fittings, welding rings, etc.

As every manufacturer of welding fittings was actively represented in the groups who set up these specifications and the corresponding dimensional specification ASA B16.9, the entire industry of welding fittings was properly covered. And since the user groups were equally represented, due consideration was given to the engineering requirements and problems.

The factor of safety was well covered in that a number of hydrostatic tests with welded assemblies were made. In practically every case the pipe which was welded to the fittings burst before the fittings, which indicated a superior fitting strength. This condition led to the paragraph which required the fittings to be at least as strong as the pipe to which it is attached.—J. H. Romann, Tube-Turns, Inc.

Readers of Heating, Piping and Air Conditioning are invited to contribute their views to these "Open for Discussion" pages, which will appear from time to time.

MUCH PLEASED WITH IT

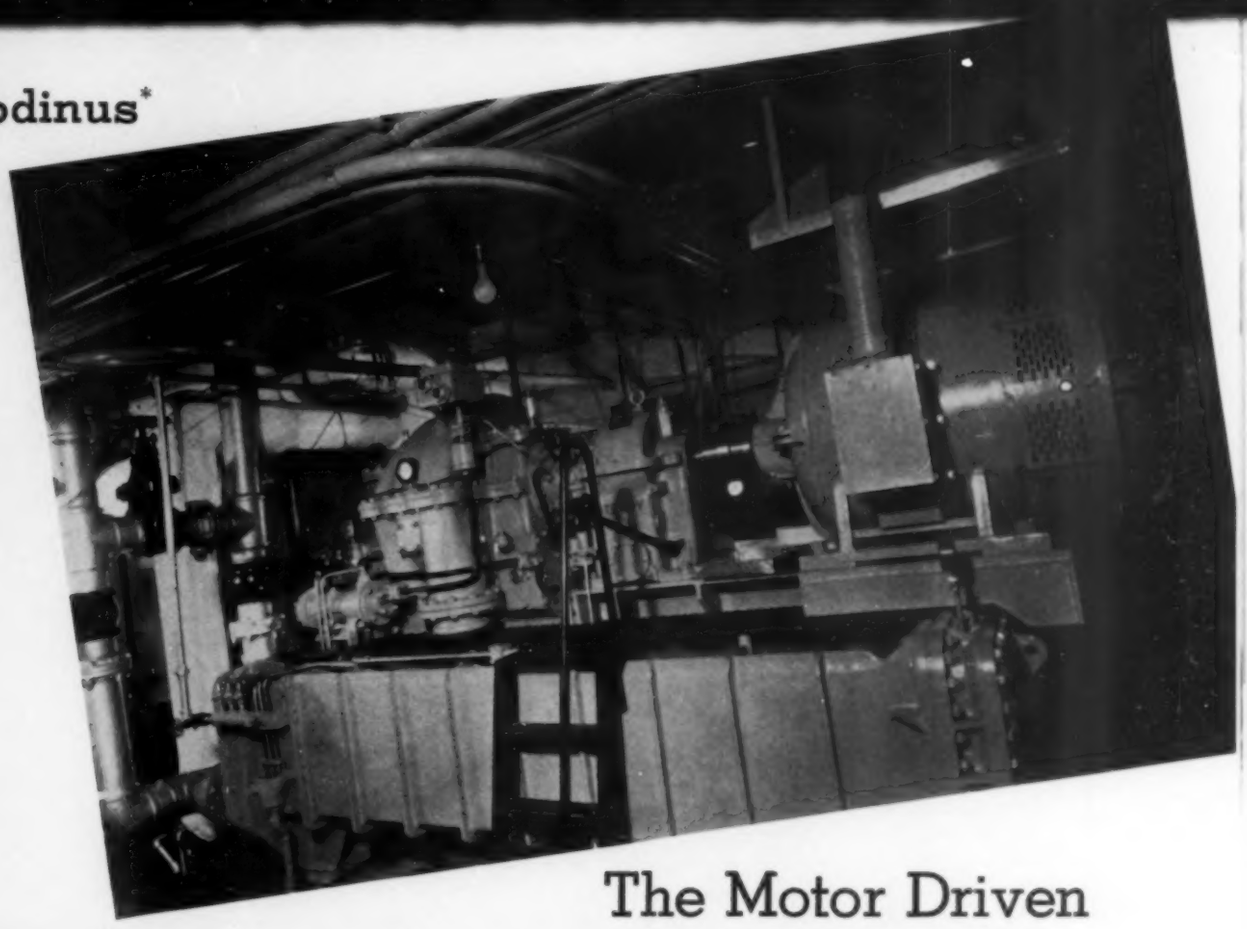
The Editor—

Several of us read Mr. Ashley's article on air change in the March HPAC and were much pleased with it. The term "air change" has been frequently misused to describe air movement within a conditioned space or as a criterion of ventilation requirements.

It is adequately and justly exposed by Mr. Ashley. Errors and consequent disappointments are bound to occur when using this crude method as a basis of design and selection of equipment. Fortunately, the engineer now has at his command more precise methods and need no longer depend on this "rule of thumb" as his guide.

Air distribution in the conditioned space has long suffered due to "air change" methods used in the selection of the distributing equipment. Mr. Ashley has shown that for this use "air change" does not accurately describe air motion in the room and may very often result in drafts; its elimination from current practice, particularly as thus used, would be highly desirable. Some of the manufacturers of air distributing equipment have long recognized the problem in connection with the selection of grilles, ceiling outlets, etc., and as a result of extended research have obtained the necessary data so that it is now possible to select with confidence, equipment suitable to meet specified requirements.—Duncan J. Stewart, Barber-Colman Co.

W. S. Bodinus*



The Motor Driven CENTRIFUGAL COMPRESSOR

CENTRIFUGALS—Thousands of tons of centrifugal refrigeration is being installed; one active field is the air conditioning of the large plants for production of defense materials, such as aircraft engines. On previous occasions, Mr. Bodinus has explained in these pages how steam turbine driven centrifugal refrigeration compressors may be used to improve the heat balance of a plant, and how turbine driven centrifugals can be installed to operate in conjunction with existing reciprocating compressors for this purpose. This month he discusses electric motor drives

SELECTION of the motor drive for a centrifugal refrigeration compressor for air conditioning use depends upon an analysis of the electrical service available, the maximum demand rate, the energy cost and the time of the demand load.

The peak air conditioning load of a restaurant, for instance, is usually at noon, while that of a theater may occur in the evening; a night club or a ballroom may have a peak load at 11:00 p.m., a department store at 3:00 p.m., a mortuary at 9:00 a.m., and a hotel at 5:00 p.m. Very little authoritative information is available as to the diversity of comfort cooling demands of a large group of customers in a given city or network area; however, a survey of this subject indicates that considerable diversity does exist, and emphasizes the need for further investigation, in order that actual diversities for this class of business may be determined.

In some sections, the refrigeration load of a comfort cooling installation is seasonally offpeak, tending to fill the summer valley in lighting and industrial power demands. In other areas, where the summer demand approaches or exceeds the winter demand, offpeak refrigeration storage offers a practical means of reducing or entirely eliminating the incremental effect of refrigeration compressor demands. Centrifugal compressors may be then installed for a capacity of 100 tons of work during the day and store up an additional 100 tons during the night with no greater demand than the horsepower requirement of 100 tons. Or, if necessity requires, the ampere input to the compressor can be very simply and constantly regulated during any hour for an amount less than the 100 tons.

A sufficient number of offpeak refrigeration storage installations, in the larger sizes, have now been made to demonstrate their practicability. Data available as to operating cost reveal that the original additional cost, if any, of a storage system will in most instances be returned within a reasonable period by the resultant savings in demand charges.

The centrifugal refrigerating compressor, particularly in large size individual units, is capable of simple control and maximum flexibility to meet the variable loads and to parallel the operation of maximum demand electrical input. Large department stores now have special summer sale days with maximum lighting demands and power demands equal to or greater than the winter demand. The motor driven centrifugal compressor unit, with its wide range of flexibility and constant temperature characteristic, can be modulated to handle a large load and yet keep the demand of electrical current where desired. The compressor is easily unloaded and can be kept just below the peak current consumption with a result of only a nominal reduction in tonnage.

*Carrier Corp.

How to Figure the Dust Concentration When Filtering Process Is Intermittent

By Frank B. Rowley* and Richard C. Jordan†

PREDICTION of dust concentrations in enclosed spaces employing continuous filtering of the air was discussed in the February HPAC, and a general dust concentration equation applicable to such conditions was derived. It is the purpose now to present similar equations for use where intermittent filtering of the air is employed.

By direct application of the general dust equation (equation 4, p. 100, February HPAC) the dust concentration at the end of a period of operation of the blower-filter system is

$$C_f' = \left[C_i - \frac{LC_o + D + FC_o(1-E)}{M+NE+Av}\right] e^{-t_1\left(\frac{M+NE+Av}{V}\right)} + \frac{LC_o + D + FC_o(1-E)}{M+NE+Av} \quad \cdots [1]$$

and the concentration at the end of a period of inactivity or "off" operation of the system is

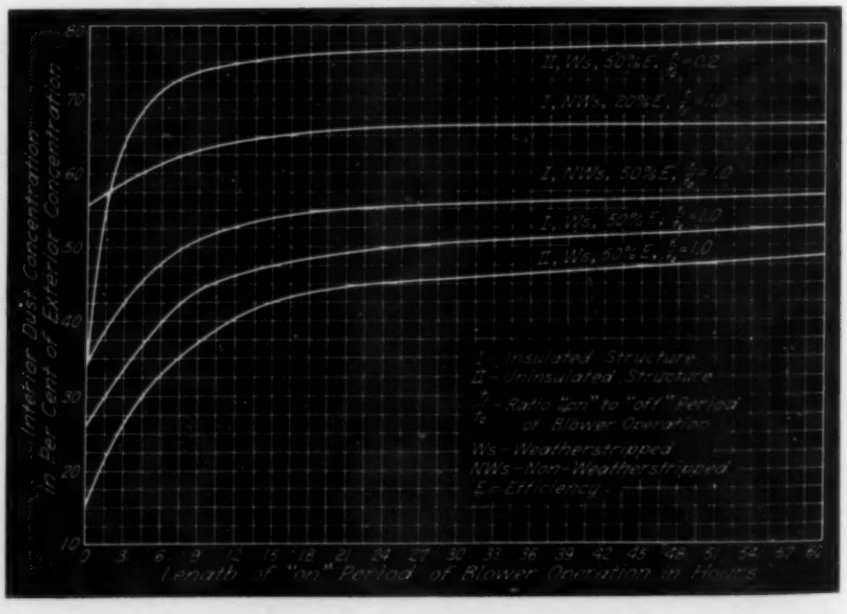
$$C_f'' = \left[C_i - \frac{L'C_o + D}{M'+Av}\right] e^{-t_2\left(\frac{M'+Av}{V}\right)} + \frac{L'C_o + D}{M'+Av} \quad \cdots\cdots [2]$$

The final dust concentration after a period of filtering or blower operation is the initial dust concentration for the following period of no blower operation, and similarly the final concentration after the period of no blower operation is the initial concentration for the next period of blower operation, etc. Thus, by successive applications of equations [1] and [2] it is possible to derive a general expression indicative of the concentration at the end of the nth cycle of operation. (Space does not permit the derivation of several of the equations presented here, but those desiring more complete information can obtain it by communicating with the authors.)

The resulting expression is, however, too complicated for practical application, and a more useful equation is one which gives the mean concentration over a complete cycle of "on" and "off" operation after equilibrium conditions have been established. Such an expression may be obtained by determining the equilibrium values for C_f' and C_f'' by allowing the number of cycles of operation to approach infinity, using these values as the upper and lower limits between which the interior concentration fluctuates after equilibrium is attained, and then integrating over one complete cycle of operation.

The expression for the resulting average concentration is still too complicated for practical application. The value of C_a, however, varies with the magnitudes of t_1 and t_2 even though the ratio of t_1 to t_2 remains constant, and it is possible to determine two simplified expressions

Fig. 1—Effect of length of filtering periods on average dust concentration. Volume of structure, 16,680 cu ft. Blower capacity, insulated construction, approximately 865 cfm. Blower capacity, uninsulated construction, 1765 cfm. Insulated construction consists of 4 in. face brick, building paper, 1 in. fir sheathing, 2 in. by 4 in. studs 16 in. on centers, 4 in. rock wool bats, vapor barrier, and wood lath and plaster. Uninsulated construction the same as insulated except rock wool is omitted


Interior Dust Concentration in Per Cent of Exterior Concentration
Length of "on" Period of Blower Operation in Hours
I, Insulated Structure
U, Uninsulated Structure
f, Ratio "on" to "off" Period of Blower Operation
Ws-Weatherstripped
NWs-Non-Weatherstripped
E-Efficiency

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for limiting cases of the expression. One of these occurs when the ratio t_1/t_2 remains constant, but the actual magnitudes of t_1 and t_2 approach infinity. The other occurs when the ratio again remains constant but the magnitudes of t_1 and t_2 approach zero. The former case results in the expression

$$C_a = \frac{\left(\frac{L'C_o + D}{M' + Av}\right) t_2 + \left(\frac{LC_o + D + FC_o(1-E)}{M + NE + Av}\right) t_1}{t_1 + t_2} \quad \cdots\cdots [3]$$

and the latter case in the expression

$$C_a = \frac{LC_o\left(\frac{t_1}{t_1+t_2}\right) + L'C_o\left(\frac{t_2}{t_1+t_2}\right) + D + FC_o(1-E)\left(\frac{t_1}{t_1+t_2}\right)}{M\left(\frac{t_1}{t_1+t_2}\right) + M'\left(\frac{t_2}{t_1+t_2}\right) + Av + NE\left(\frac{t_1}{t_1+t_2}\right)} \quad \cdots [4]$$

The physical significance of these equations is as follows. Equation [3] approximates the average concentration in an enclosed space after equilibrium conditions have been reached and when the periods of operation and inoperation of the blower, t_1 and t_2, are very long. This equation should be used only when these periods are a number of hours in length. Equation [4] approximates the average concentration after equilibrium when the periods of operation, t_1 and t_2, are short. This equation results in a reasonably accurate approximation when the periods are one or two hours in length or shorter.

That equation [4] is a close approximation to the average equilibrium dust concentration for most practical intermittent operating conditions is indicated by Fig. 1, which shows the relationship between the length of the "on" periods of blower operation and the average equilibrium dust concentrations as determined by equation [4] for five practical installations. These curves are based upon the same structure but with variations in insulation, weatherstripping, filtering efficiency and ratio of lengths of "on" to "off" periods of operation.

Although all of the curves shown in Fig. 1 rise rapidly at the start and then level off as the periods of filtering lengthen, it should be noted that the lengths of the periods are given in hours while under most practical operating conditions the lengths are in minutes. The curve showing the most rapid increase in concentrations with lengthening of the time period is that for the uninsulated construction with a ratio of t_1/t_2 of 0.2. This would naturally show a more rapid rise as the "off" periods are five times as long and the elapsed time per cycle three times as long as in the case of any of the other curves.

By an analysis of these curves and the original equations it can be seen that, if the periods of "on" and "off" blower operation are less than an hour in length, there are only slight differences between the results obtained by either the complete expression for the resulting average dust concentration (not given here) or equation [4]. As an illustration, for the insulated construction, weatherstripped, filtering efficiency 50 per cent and t_1/t_2 equal to 1.0, the average concentration increases from 26.2 to 27.1 per cent when the periods of operation are increased from infinitesimally short to one hour in length. In the case of the uninsulated construction, weatherstripped, filtering efficiency 50 per cent, and t_1/t_2 equal to 0.2, the average concentration increases from 34.5 to 35.5 per cent when the periods of "off" operation are increased from a value approaching zero to one 50 min in length. Similar analyses hold for the other cases. Therefore, it is safe to conclude that for most filtering installations where the periods of operation are shorter than one hour in duration, the average dust concentration under equilibrium conditions may be approximated by means of equation [4].

It is interesting to investigate the effect of variations in filtering efficiency upon the interior dust concentrations by analysis of equation [4] for intermittent operation and the following equation for *continuous* filtering operation, which was given on p. 100 of the February HPAC:

$$C_r = \frac{LC_o + D + FC_o(1-E)}{M + NE + Av}$$

In either case if we consider C_r and E as the only vari-

SUMMARY—The authors have made studies on atmospheric dust and methods of measuring it for several years, but until now nothing of a semi-popular nature about the various factors involved, or the developments which have been taking place in this field, has been published. It is the authors' feeling that the question of atmospheric dust has been more or less neglected insofar as methods of measurement or standards are concerned—yet it is most important in many air conditioning problems, particularly now that industrial activity is increasing and will continue to do so under the requirements of our national defense program. . . . . The factors affecting dust concentration levels in enclosed spaces were explained in the December HPAC; it was pointed out that the rates at which dust laden air is introduced into the space, with which dust is produced in the space and with which dust settles from the air, and the rate at which air is filtered, are usually of greater importance in the determination of actual dust concentrations than the efficiencies of the filters themselves. In February, the application of the data to continuous filtering processes was given, and this month intermittent operation is covered. . . . . The theory upon which this discussion is based was proposed by Dr. Jordan

Symbols Defined

t_1 = length of "on" period of operation, minutes;
t_2 = length of "off" period of operation, minutes;
L = infiltration air when blower operating, cfm;
L' = infiltration air when blower not operating, cfm;
M = exfiltration air when blower operating, cfm;
M' = exfiltration air when blower not operating, cfm;
N = recirculated air, cfm;
F = outside air, cfm;
C_a = average dust concentration, particles per cu ft;
C_r = inside air dust concentration, particles per cu ft;
C_r' = dust concentration at end of "on" period of operation, particles per cu ft;
C_r'' = dust concentration at end of "off" period of operation, particles per cu ft;
C_i = initial inside concentration at beginning of operation period, particles per cu ft;
C_o = outside air dust concentration, particles per cu ft;
E = filtering efficiency, per cent;
D = dust produced from inside sources, particles per min;
V = net volume of room, cu ft;
v = effective settling velocity of dust, fpm;
A = floor area of enclosed space, sq ft;
B, G, P = constants;
e = natural logarithm base (2.718).

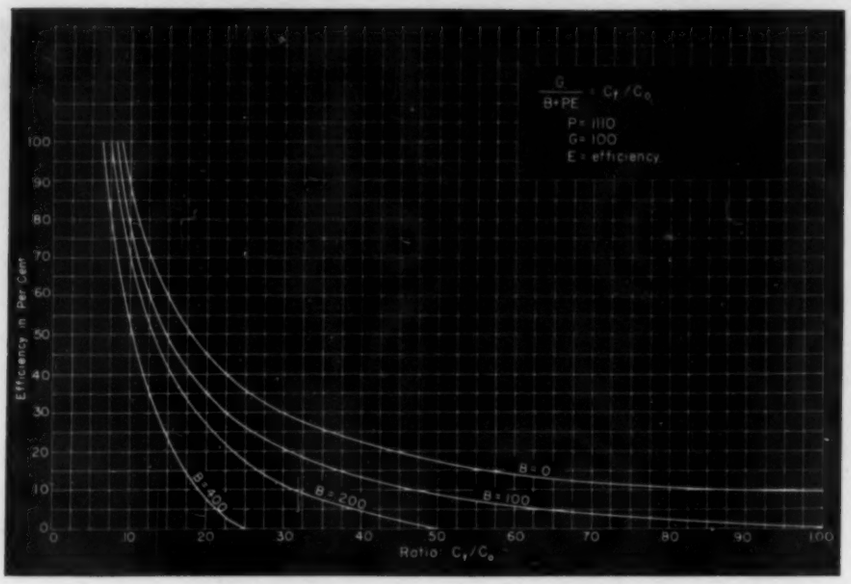


Fig. 2—Effect of filtering efficiency on the ratio of interior to exterior dust concentration C_t/C_o

bles, let F and D equal zero to simplify analysis, and combine the constants, the resulting relationship is

$$C_t/C_o = \frac{G}{B + PE}$$

where G, B and P are constants. If the value of B is small in comparison with PE and can be disregarded, the equation takes the form

$$C_t/C_o = \frac{G}{PE}$$

This will be recognized as the equation of a rectangular hyperbola and represents the conditions sometimes approached in practice when the rate of filtering is high. For most installations B is in the order of magnitude of G, whereas PE may be considerably larger.

Fig. 2 shows the graphical relationship between efficiency E and the ratio C_t/C_o for values of B ranging from 0 to 400. In determining these curves a value of 100 has been assumed for A and a value of 1110 for P. It should be observed that increases in filtering efficiency above low initial efficiencies result in much more marked decreases in the interior dust concentration than do like increases in efficiency above high initial values. For example, when $B = 100$, increasing the efficiency from 0 to 10 per cent decreased the ratio of C_t/C_o 52.5 percentage units, whereas increasing the efficiency from 70 to 100 per cent, a three times greater rise in efficiency, decreased the ratio of C_t/C_o only about three percentage units. It would therefore appear that in many cases too much emphasis had been placed upon the efficiency characteristics of filters when the differences are in the higher values, and that insufficient weight has been given to the other important properties, such as life and dust holding capacity.

[This is the third of the Rowley-Jordan articles on predicting dust concentration. The first, which appeared in December, 1940, was devoted to factors affecting dust concentration levels in enclosed spaces. The second, in February, 1941, explained how to figure concentration when the filtering process is continuous. A fourth article will give typical examples of concentration analyses.]

INSTRUMENTS AND CONTROLS INCREASE OUTPUT

No two coals, and certainly no two different fuels, are sufficiently alike to allow definite promises of savings. The following improvement would result in many coal fired boilers; similar savings would apply to burning other fuels: If a boiler were being operated to burn coal in such a way that an 8 per cent CO_2 existed in the boiler exit gas, that would mean a preventable loss (not a total loss) of $115 for each $1000 of coal burned. Burning this same fuel with a 14 per cent CO_2 flue gas would reduce this preventable loss to about $15. This is a saving of $100 for each $1000 worth of coal burned. This 10 per cent is a lot of dollars in the coal bill for any boiler plant, large or small. Various boilers have been found operating at even less efficient conditions than represented by 8 per cent CO_2. Smaller increases in efficiency of burning result in correspondingly smaller savings. A combustion meter could thus pay for itself in quick time.

While steam flow meters may not point out as large possible savings as will a combustion meter, the steam flow meter is indispensable as it provides a continuous indication to the operator, plus a permanent record, of exactly the amounts of steam generated at all times, and it furnishes an accurate total of the steam generated over any definite period. Some of the most important steam flows to be metered are to steam turbines, bleed steam from turbine to each intermediate pressure header, exhaust steam for heating water and plant heating plus steam for process. Feedwater is always measured and is frequently subdivided into return condensate and make up water.

Distribution meters usually increase the capacity of existing boilers in a different way. They put the finger on unnecessary uses and excessive waste of steam after it leaves the boiler room and cause these wastes to be eliminated. This additional boiler capacity is then available for actual steam needs rather than for supplying wasted steam. The ancient—and still used—practice of charging fixed percentages of the total steam cost to certain departments offers no help in spotting the dollar leaks. It gives the head of each department no incentive for eliminating these leaks. It is human nature to blame the excessive use of steam, water, heat and light on the other departments when it is admittedly an arbitrary spreading of costs. When accurate meters are installed on each of these lines, there is no way of dodging the fact that a particular department used a definite number of dollars worth.—CHARLES W. PARSONS, Republic Flow Meters Co., at the Midwest Power Conference.

4000 Ton Decentralized Air Conditioning System for Big Black-Out Plant

69 Separate Systems for Douglas Aircraft Project

By .... H. M. Hendrickson*

AT LAKEWOOD BLVD. and Carson St., in North Long Beach, Calif., the enormous new black-out assembly plant of the Douglas Aircraft Co. is rapidly taking form. Here we find the unique situation of two competitors collaborating in the interest of national defense to install industrial air conditioning for the factory buildings and comfort air conditioning for the administrative offices in record breaking time.

High Internal Load

With the mild climate near the California coast (the ASHVE Guide gives summer design conditions of 90 F dry bulb and 70 F wet bulb for the adjacent city of Los Angeles), one might think that air conditioning would be unnecessary. However, the extremely high internal load from the men at work, machinery, motors, lighting, processes, etc., coupled with the lack of windows and the high sun load on the dull black surfaces of the factory buildings, make necessary 4000 tons of refrigerating capacity to cool the buildings to 80 F dry bulb and 48 per cent relative humidity for manufacturing and to 78 F dry bulb and 50 per cent relative humidity in the offices.

Separate Systems

On the Douglas Aircraft project we find no central heating and cooling plant which one well-directed bomb might put out of commission, thus crippling the entire plant. Here the unitary idea is carried out to a remarkable degree without penalty in first cost over the central station type of plant. There are 69 completely separate refrigerating systems for air conditioning, each averaging about 60 tons and each having its own condensing unit, economizer, direct expansion "Freon" cooling coils, steam heating coils, fan, filters and duct system.

Equipment Location

Each factory building has two or more independent boiler plants for heating and from eight to 15 independent refrigerating and fan systems for cooling. The refrigerating and fan units are situated in the trusses directly under the roof, as the entire space from the bottom chord of the truss to the floor must be kept clear for the operation of cranes. All the ductwork from each fan unit is located above the bottom chord of the truss with air supply outlets in each bay. For even temperature conditions when either heating or cooling, return ducts are used with drops at columns and along outside walls to within 6 ft of the floor. Steam booster heaters are employed in the ducts supplying all outside exposures. Each fan station handles approximately 36,000 cfm.

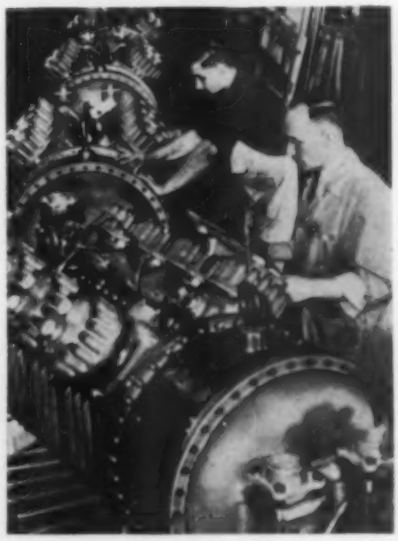
Six Main Buildings

The six main manufacturing buildings are constructed of corrugated iron with 1 in. of structural insulation on the inside. The roof, of wood and composition roofing, is similarly insulated. The two largest assembly buildings each contain 270,000 sq ft of floor space with 15 fan and cooling stations; the two manufacturing buildings each contain 195,000 sq ft of floor space with 12 fan and cooling stations; and the two smaller stock and manufacturing buildings each contain 144,000 sq ft of floor space with eight fan stations. Only one of the latter has refrigeration, the stock building being ventilated only.

New Idea

Thus, we have here another new idea in industrial air conditioning—the entirely decentralized plant. Its advantage in war times cannot be over-estimated as one or more buildings or parts of buildings could be blown up, but the rest of the plant could still be operated with the unitary equipment, and the duplicate and standby services of gas, electricity, water, etc.

These compact air conditioning compressors will provide cool air for workers in two defense black-out plants. Built at Springfield, Mass., they are being checked for shipment to the West and South. Summer heat in windowless factories may push the temperature up to 110 deg, but air conditioning will maintain comfortable and efficient temperatures, says P. Y. Danley, manager of the Westinghouse air conditioning department



*Assistant branch engineer, York Ice Machinery Corp.

NARROW SLOTS — Little information is available on slotted air discharge systems, though this method of air distribution has considerable use in industrial air conditioning work. These jobs are usually designed by rule of thumb, with plenty of provisions for balancing and changing the air distribution after the system is put in operation. . . . . The author was called upon to design a slotted air discharge system and decided to secure first-hand information in the field. He has summarized the results of his "survey" here, and he will continue the discussion further in a second article

The slot discharge type of air distribution used in an orange precooling room. The supply duct is in the center with the return ducts at each side

AIR DISCHARGE from Narrow Slots

Results of Field Observations and Experience

Summarized by F. F. Stevenson

RECENTLY I was asked to design a slotted air discharge system for industrial air conditioning and found it necessary to get some first-hand information about this method of air distribution. I therefore inspected several installations.

These jobs all seem to require considerable attention after being completed in order that they may function properly. Such additions as "blank offs," baffles and other impedimenta were not the exception, but the rule, before the air flow was balanced and drafts and dead spots eliminated. Several ventilating engineers were consulted, but there seemed to be a considerable difference of opinion as to the proper design of such a system. The information given here is not the result of laboratory tests, for so far as I was able to find out, there are no such data available. It is based on field observations and experience.

Slot Construction

Since the advent of the aspiration type of ceiling diffusers and streamlined discharge grilles, the narrow slot air discharge has little place in comfort air conditioning, but in industrial applications it has and will probably continue to have a wide application. It has the virtue of low cost and will project a continuous thin stream of air into the space to be conditioned. If properly constructed, there should be no dead spots or excessive drafts. Also, and this deserves the attention of the designing engineer, it will distribute substantial volumes of air with high discharge velocity at the slot, with a very short throw. The advantage of this in many situations is obvious.

A simple and low cost method of constructing a slot air discharge is to use a supply duct of constant cross sectional area to feed the slot. The slot may be placed along only one side of a room or space, terminating in a dead end, or along several sides, or entirely around the space with no dead end. In industrial systems, the duct usually is exposed, with the slot cut directly into it, but in some cases, the duct is built back of the wall or partition with a throat leading from the duct to the slot. The throat varies in depth from 2 to 3 in. up to 1 ft. The slot is placed fairly high up in the wall and usually the air discharge is horizontal. With exposed ducts, the slot may be placed in the center of the side of the duct, with a horizontal throw, or in the lower end of the side, also with a horizontal throw. There are also instances where the

slot is cut into the bottom of the duct with a vertical downward air discharge. In industrial work, we frequently see the exposed duct, with a slot cut in each side, discharging the air horizontally in two directions. A return duct with an exhaust slot may be built in conjunction with the supply slot.

I encountered one installation where the air was fed from the main supply duct (back of the wall) through a number of take offs, each one feeding a separate slot section. The slot was continuous; that is, where one section stopped another began, but each section was separate from the other and had its own air supply. This system probably was more expensive to build than one with a supply duct of constant cross sectional area and one long continuous slot, but its air supply can perhaps be controlled and balanced more readily.

In some instances turning vanes are used to direct the flow of air outward through the slot utilizing the velocity pressure of the air. Turning vanes consist of curved metal strips so placed in the duct as to change the direction of the air. Another form is the turning core, placed at the slot, serving the same purpose. These devices help straighten out the air stream and some of the duct velocity pressure is used for discharge purposes. As velocity diminishes, however, the utilization of velocity pressure is so small that it is negligible for practical calculations.

The height of the slot may vary from ¾ in. up to around 3 in., with the throw of air a function of the square root of slot area, the discharge velocity and the aspect ratio of the air stream, width to depth.

It has been stated that the aspect ratio of air supply grilles of average proportions has little effect on their throw, but the aspect ratio undoubtedly has a major effect on the throw of air from narrow slots.

Coefficient of Discharge

There is considerable variation in the coefficient of discharge from the slots. It has been found that the coefficient is higher when the slot has a throat back of it. The effect of the throat is to reduce turbulence and straighten out the air stream, both of which make for a more effective discharge. The height and contour of the slot also affect the coefficient. With various types and sizes of orifices, with and without throats, the coefficient of discharge has been found to range from 0.55 to 0.85. With a well constructed slot and a throat of around 5 to 8 in. deep, a coefficient of from 0.75 to 0.82 will probably be average. Without a throat, the coefficient may be assumed to run from 0.58 to 0.70.

When constructing the throat, care should be exercised lest it sag at one end, because if the approach air stream passing through the throat is not on a directly horizontal plane, the throw of air will be inclined upward or downward. I know of one installation with the slot fairly close to the ceiling line, where an upward spring of the throat impinges the air on the ceiling with a serious smudging effect as the result.

Air Throw

The air throw from a narrow slot, considering the discharge velocity, is very short. This is undoubtedly due to the shape of the air stream, which has considerable width and very little thickness. This has the effect of presenting a large area, as compared with the air mass, against which the air in the space interposes resistance, reducing the velocity and travel of the discharge air. This factor makes the slot discharge satisfactory for applications where considerable amount of air is required, but where a short throw is essential.

Pressures and Velocities

The pressures and velocities in the supply duct are most important and must be carefully considered before the plans are approved. They are the factors which cause large variations in the air distribution and if they are not calculated correctly, the job of balancing the system may be greater than the actual construction.

The supply duct, of constant cross sectional area, is, in effect, a sort of plenum, supplying air under pressure to the slot. As the discharge of air through the slot (assuming either a horizontal or vertical discharge) is at a 90 deg angle to its flow through the duct, the velocity pressure through the duct at any point cannot be too high as compared to the static pressure. If we have a combination of high velocity and low static pressures, the tendency of the air stream, due to inertia, is to keep on going through the duct instead of flowing out through the slot.

Naturally, the pressure loss of discharge also has an influence in this regard and many systems are designed so that the height of the slot may be varied and the system balanced after it is in operation. In some cases the area of the slot will be less at the beginning of the run and greater at the far end, and in others the reverse is true. With a duct of constant cross sectional area and with a constant supply of air entering at one end, the air volume is constantly being diminished because of its discharging through the slot and it also has a constantly diminishing velocity. Toward the latter portion of the run, the air velocity is very low, as is its volume. When no turning vanes are used we depend on static pressure not only to turn the air stream at right angles, but also to supply the energy necessary for its discharge.

The flow of air through a run of straight duct, serving the slot, encounters three sources of pressure loss: (1) Frictional resistance of the duct walls. (2) Loss caused by the turn in direction and turbulent flow toward the slot. (3) Shock loss caused by changes in velocity of the air flowing through the duct.

With the velocities maintained in the ducts in normal practice, none of these losses are high. The friction loss may be determined as a matter of academic interest, although with good construction it should not exceed 0.03 in. of water and in most cases is considerably less. The second loss is difficult to measure and in the latter portion of the duct it is very small. This loss is really a part of the discharge loss and is so considered here. The shock loss may be dismissed as negligible.

In calculating the friction loss, it is well to remember that with constantly decreasing volume and velocity of air through the duct, we have a constantly decreasing loss per foot of run.

Friction Loss Formula

I was given the following empirical formula for determining the friction loss through a run of straight duct:

$P = P_1 \times$ length of duct in ft $\times$ 0.25

where P = friction pressure loss of duct, in. of water;

P_1 = friction pressure loss per ft at entrance of duct, based on cross sectional area and velocity at this point, in. of water;

0.25 = an empirical constant.

As the friction loss varies with the square of the air velocity, it is obvious that this empirical formula for the calculation of friction loss of air with a constantly diminishing velocity will not give accurate results for the losses at intermediate points along the run of duct. However, for this work it is necessary only to calculate the loss for the entire run.

This formula was checked by calculating the loss of several such ducts foot by foot and adding the results. For runs of considerable length, say 75 to 125 ft, it gives good results. For smaller runs, say from 25 to 50 ft, the constant should be changed from 0.25 to 0.35. This is of course for straight ducts.

If there are elbows, the dynamic losses therefrom can easily be added to the friction loss. In ascertaining the dynamic loss of an elbow, a good approximation of the air velocity at the elbow is necessary.

Some designers have set up a rule of thumb that the duct entrance velocity should be not more than two-thirds of the slot discharge velocity. This usually means a large duct and I found that the rule was not always followed in actual practice.

[A second article by Mr. Stevenson will discuss static pressures and other details of the design of narrow slot air discharge systems.]

ASTM COMMITTEES TELL OF PROGRESS

At its meeting during ASTM committee week held in Washington recently, ASTM's committee C-16 on thermal insulating materials gave consideration to the expansion in its scope of activity to cover thermal insulating materials in the building construction field. As a result of this addition to its field of endeavor, the committee plans to enlarge its membership to include those having an interest in this material. Means of initiating work on this subject were discussed.

A subcommittee, under the chairmanship of R. H. Heilman, Mellon Institute of Industrial Research, has been studying a number of methods in current use for determining the physical properties of preformed insulation. As a result of this work, proposed standard methods have been completed for determining the crushing strength and flexural strength of preformed block thermal insulation. Proposed definitions for preformed block insulation and for thermal insulating cement will be submitted to the ASTM for publication as tentative.

The subcommittee which deals with studies of the physical properties of plastic insulation, under the chairmanship of H. H. Rinehart, Johns-Manville Corp., reported the completion of several new methods. These cover a detailed procedure for mixing thermal insulating cements and complete methods of test for covering capacity and volumetric change upon drying for thermal insulating cements, and a test for bulk density of thermal insulating cement.

The subcommittee dealing with blanket, flexible, loose fill and miscellaneous insulation, under the chairmanship of R. E. Cryor, Union Asbestos and Rubber Co., presented to the committee a new method of test for determining the thickness and density of blanket type thermal insulating materials.

The joint committee on thermal conductivity of all forms of insulation sponsored by four organizations (the American Society of Heating and Ventilating Engineers, the American Society of Refrigerating Engineers, the National Research Council, and the American Society for Testing Materials) has developed a method of test for the determination of thermal conductivity by use of the guarded hot plate method. Further study is to be given this method by the joint committee before submitting it to the sponsoring organizations.

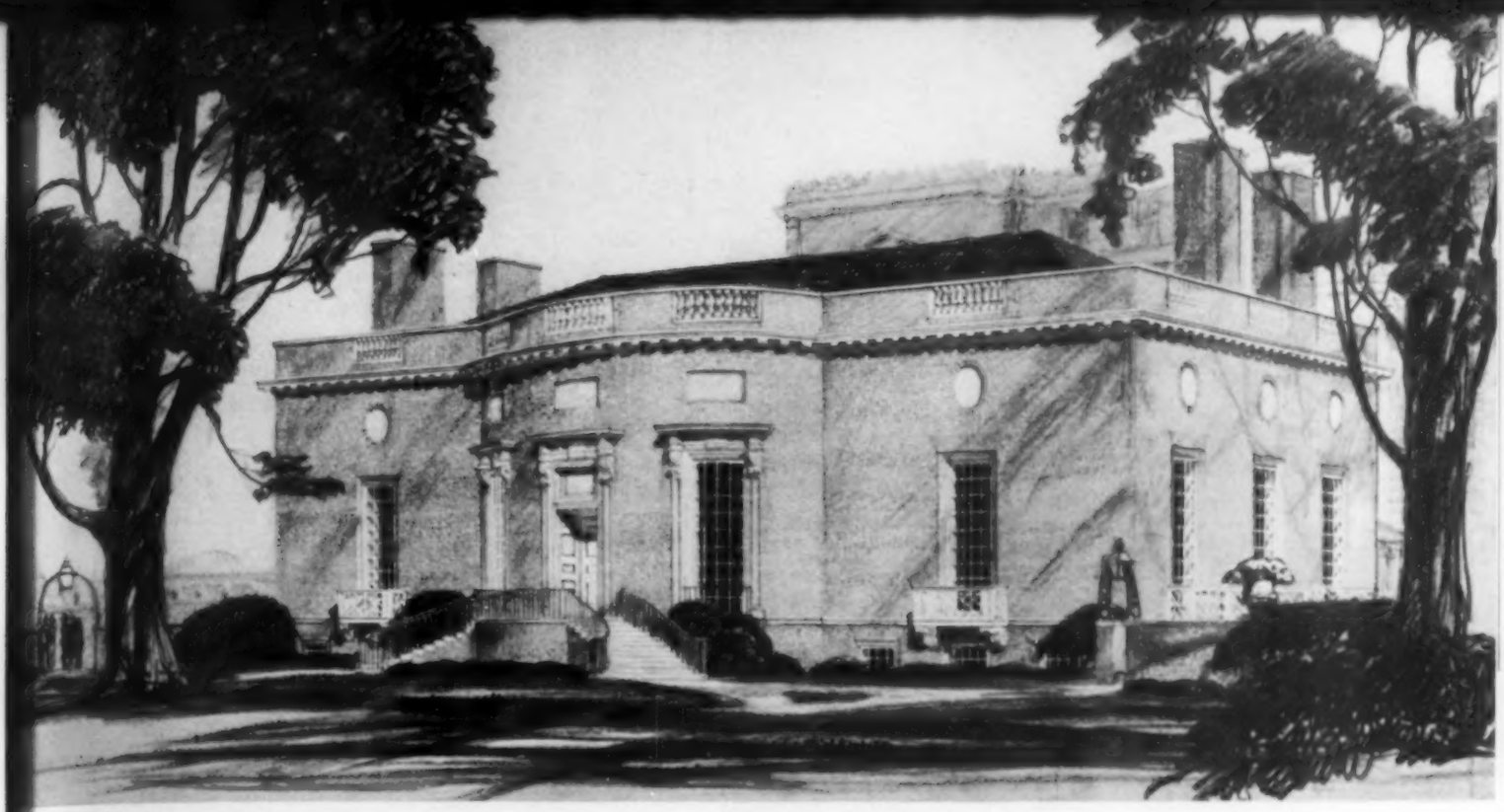
Officers of committee C-16 on thermal insulating materials are chairman, J. H. Walker, engineer assistant to general manager, The Detroit Edison Co. and secretary, E. T. Cope, The Detroit Edison Co.

An extensive meeting of the subcommittee on steel tubing and pipe, T. G. Stitt, chief inspecting engineer, Pittsburgh Steel Co., chairman, resulted in a number of recommendations. One is to delete all references to temperature ranges in the standard specifications for electric resistance welded steel pipe (A 135-34) which now limits the material to top service of 450 F. Another recommendation is to set up requirements for copper molybdenum iron boiler tubes. This material eventually is to be considered for incorporation in the existing basic standard A 83-40 which covers lap welded and seamless steel and lap welded iron boiler tubes.

At a meeting of committee D-5 on coal and coke, action was taken to recommend the adoption as standard of the present tentative method of test for index of dustiness of coal and coke (D 547-39 T). This method, first published in tentative form by the ASTM in 1939, measures the relative dustiness of coal and coke when handled. It has proved of especial value in the evaluation of various processes for treating coal and coke in the production of so-called "dustless fuel."

REDUCES ERRORS

Fewer banking errors, a definite pick-up in personnel efficiency and a more pleasant relationship with customers and depositors have resulted from the year 'round air conditioning of the Security National Bank at Greensboro, N. C. All three floors of the building are air conditioned. The trust department in the basement receives the same cool, dry air supplied to the main banking rooms on the first floor and the directors' room and general and private offices on the second floor, regardless of the outside weather. Extreme dampness in wet weather, and excessive heat in hot summer weather, had been major problems formerly.—Edward T. Murphy, vice-president, Carrier Corp.



Air Conditioning for Protection

G. K. Saurwein, Superintendent of Engineering, Harvard University, Describes New Rare Book Library . . . . . Controlled Atmosphere Will Protect Valuable Collection from Deterioration

DURING the 300-odd years of its existence, Harvard university has acquired by purchase and by gift a unique collection of rare books and manuscripts. The material dates from the beginning of printing down to recent times, includes 2000 books from the 15th century, manuscripts from much earlier times. The rarer pieces of the collection have been grouped for separate housing and include about 100,000 volumes of printed books and unnumbered thousands of manuscripts The collection is increasing year by year as bequests are made to the university and the library rounds out by purchase the various parts of the collection.

In its entirety, this is one of the most important collections of printed material in the country. A number of special groups in the collection are, in their particular fields, the most complete in the world. It is the size and completeness of these rare book collections as well as of its general collection that places the Harvard library in the front rank of university libraries.

PROTECTION—Air conditioning is used for many purposes—processing, comfort, efficiency, health (to name but a few). At Harvard university's new rare book library, Cambridge, Mass., it will protect a collection of books and manuscripts worth millions of dollars from deterioration due to varying temperatures and humidities, dust and sulphur dioxide in the air. . . . The air conditioner — the cabinet for which is built using a new method—comprises an electrostatic air filter, preheating coil, spray chamber with a battery of misting nozzles, heating coils, direct expansion cooling coils, and a bank of sulphur dioxide and odor adsorbing canisters. The automatic controls are of the pneumatic type. . . . The lighting in the building will be somewhat unusual, fluorescent lamps being used extensively. All stacks and work spaces will be fitted with fluorescent fixtures, with louvers designed to permit a small amount of light to pass upward to relieve the contrast between the lighted areas of the room and the ceiling. . . . An architect's drawing of the new Harvard library is shown above

Best Possible Care

Such material, the value of which runs into many millions of dollars and much of which is irreplaceable, must have the best possible housing and care. The paper, leather, parchment and binding materials of these rare books would not endure long if subjected to the deteriorating influences of sunlight, sulphur dioxide in air, too great variations in temperature and humidity, dust abrasion in cleaning, etc.

To extend the useful life of this rare material as far as possible, a building is being erected which will be fully air conditioned and in which the books and manuscripts will be protected against deteriorating influences. This unit—to house about

250,000 volumes—is about 60 ft by 120 ft in area, with three floors above and three floors below grade. The building is connected to the main building (the Widener library) by a bridge at the second floor and by a tunnel at the sub-basement level. The air conditioning apparatus will be installed in the basement of the main building. Supply and return air ducts, heating water piping, service water piping, compressed air lines, electrical mains, etc., will be carried into the new building through the tunnel. Thus, any equipment hazards will be removed from the new building.

The concrete walls below grade and the brick walls above grade will be waterproofed with bitumastic material and insulated with 1 in. cork; the roof will be insulated with glass wool. Windows will be fixed and double glazed, with a 3 in. air space between the glazing.

Heating and Ventilation

The heating system will be of the split type, with enclosed hot water cast iron convectors to overcome the heat loss at the windows, and warm air for general heating and for ventilation. According to the design, the supply duct after leaving the conditioner is divided into two ducts, one serving the floors above grade and the other those below grade. The latter is fitted with a booster heater, for in certain seasons it will be necessary to supply some additional heat to the lower floors. The regular booster heater is supplemented by a finned type, all copper electric heater with external terminals to function during summer months when the steam system is shut down, and for dehumidification.

The warm air for some rooms is admitted through adjustable grilles near the ceiling, and for others through circular ceiling registers. The return air is taken through registers near the floor in the usual manner. Air from toilets, closets and the like will be exhausted to atmosphere. Fireplaces are fitted with induced draft fans. Branch ducts are fitted with volume control dampers with indicating and locking quadrants. Behind each air supply register is a friction type damper accessible for adjustment. The duct work is made up from galvanized iron sheet metal, with fusible link, self closing fire dampers as recommended by fire protection authorities.

The supply and return air ducts in the tunnel will be lined for a distance of 20 ft with sound absorption material. Some of the inlet register boxes and ducts are lined to a distance of 15 ft with ½ in. sound absorption material to guard against transmission of noise between adjacent rooms supplied from the same main duct.

The standard weight steel hot water heating piping will be welded, with steel socket type streamlined fittings.

The Conditioner

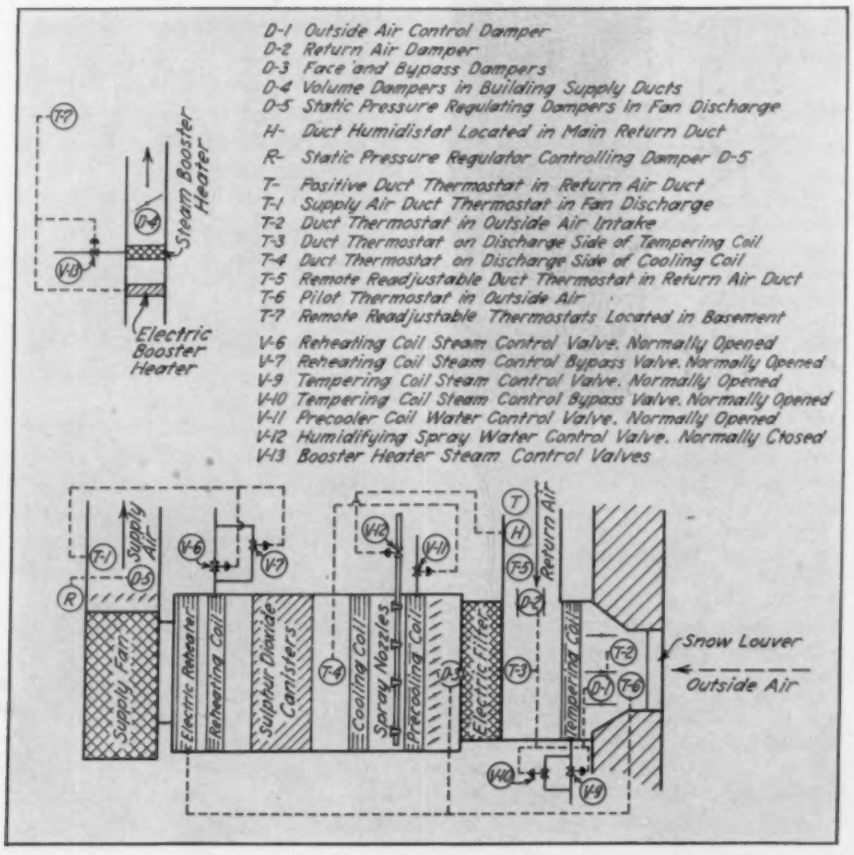
The conditioner cabinet is mounted on a concrete base, depressed in the center to serve also as a drip pan.

The unit, shown diagrammatically in the sketch, is made up of a self cleaning electrostatic air filter, precooling coil for 55 F well water, preheating coil, spray chamber with a battery of misting nozzles, heating coils, direct expansion cooling coils, and a bank of activated carbon SO_2 and odor adsorbing canisters. Coils will be of the porcupine quill type with internal spirals.

The cabinet will be fabricated by means of a type of construction used hitherto chiefly for refrigerator cabinets and truck bodies. The standard shapes used in this structure lend themselves nicely to the assembly of insulated conditioner cabinets at low cost. Using this method, a cabinet is made up of special galvanized rolled sections and panels which combine great strength and lightness due to the fact that the panels are put in tension in all directions by special frame members fitted with tension bars drawn into place with flush set screws. The structure is readily disassembled without damage to parts.

Well water at 55 F greatly reduces not only the city water bill but also the capacity of the refrigeration machine. The cold well water will provide half the required cooling capacity. The 30 ton compressor, using "Freon" refrigerant, has a two step control. The direct expansion coils have clean-out openings conveniently arranged for servicing.

Schematic diagram of air conditioning system


D-1 Outside Air Control Damper
D-2 Return Air Damper
D-3 Face and Bypass Dampers
D-4 Volume Dampers in Building Supply Ducts
D-5 Static Pressure Regulating Dampers in Fan Discharge
H- Duct Humidistat Located in Main Return Duct
R- Static Pressure Regulator Controlling Damper D-5
T- Positive Duct Thermostat in Return Air Duct
T-1 Supply Air Duct Thermostat in Fan Discharge
T-2 Duct Thermostat in Outside Air Intake
T-3 Duct Thermostat on Discharge Side of Tempering Coil
T-4 Duct Thermostat on Discharge Side of Cooling Coil
T-5 Remote Readjustable Duct Thermostat in Return Air Duct
T-6 Pilot Thermostat in Outside Air
T-7 Remote Readjustable Thermostats Located in Basement
V-6 Reheating Coil Steam Control Valve, Normally Opened
V-7 Reheating Coil Steam Control Bypass Valve, Normally Opened
V-9 Tempering Coil Steam Control Valve, Normally Opened
V-10 Tempering Coil Steam Control Bypass Valve, Normally Opened
V-11 Precooler Coil Water Control Valve, Normally Opened
V-12 Humidifying Spray Water Control Valve, Normally Closed
V-13 Booster Heater Steam Control Valves
Steam Booster Heater
Electric Booster Heater
Supply Air
Return Air
Snow Louver
Outside Air
Supply Fan
Electric Reheater
Reheating Coil
Sulphur Dioxide Canisters
Cooling Coil
Spray Nozzles
Precooling Coil
Electric Filter
Tempering Coil

The air conditioning system was designed to maintain winter conditions of 70 to 72 F dry bulb, and 45 per cent relative humidity with a maximum variation of 5 per cent above or below; summer conditions are 78 to 80 F maximum dry bulb and the same relative humidity as in winter.

Controls

The automatic temperature and humidity controls are of the pneumatic type. Most of the rooms have thermostats controlling the pneumatic convector valves, the seats of which will not be drilled. There are also controls on some of the supply air dampers.

A dew point thermostat installed on the leaving side of the direct expansion cooling coil will operate, through electro-pneumatic switches, the unloader device, the starter of the compressor, and the precooler coil water valve in such a manner that the compressor control is inoperative until the water valve is wide open.

A duct humidistat, situated in the return air, will control the water valve on the humidifier sprays in winter and the precooler water valve and compressor control for summer dehumidification.

A remote readjustable duct thermostat controlled by the outside air pilot thermostat will control the return air and face and bypass dampers in summer.

A positive acting return air thermostat will automatically change over the controls from winter to summer operation when the building temperature reaches a predetermined point.

A static pressure regulator in the fan discharge duct will control the air volume damper.

Two hot water regulators for converters with normally closed valves will be reset by the outdoor thermostat.

Mercury type thermometers are installed adjacent to each duct thermostat.

Perry, Shaw & Hepburn are architects for the new building, and the George A. Fuller Co. is the general contractor. Buerkel & Co. are the heating engineers. The work is being carried on under the direction of the staff of the business manager of the university coöperating with K. D. Metcalf, director of the university libraries, and W. A. Jackson, assistant librarian in charge of rare books.

IMPROVES WELDING FACILITIES

Plant expansion and the resulting relocation of several production departments at the Reliance Electric & Engineering Co., Cleveland, Ohio, made it possible to introduce some improvements in the facilities afforded the men who handle the welding operations.

The first improvement was concerned with introduction of canvas enclosures for the welding booths. The only rigid part of the enclosure now is the plant wall against which all the booths are located. Three sides of each booth make use of regular brown welding canvas, hung by the corners from standard safety zone stanchions, 6 ft 2 in. in height, with a base 125 lb in weight. Each stanchion has bolted to it a light sheet steel strip which projects approximately 12 in. on either side of the upright to prevent injurious light rays from passing through the curtain junctions, and to provide a snug enclosure of each booth. The stanchions used at the corners of the booths have two of these light sheet steel strips bolted at right angles to each other to form tight corners. Eyelets in the top corners of each section of welding canvas permit fastening it securely to hooks in the stanchion tops. This method of holding the enclosures in place eliminates wires or pipe supports across the tops of the booths, permitting greater freedom and safety in depositing work from cranes.

Absence of rigid frames of any sort around welding booths or across their tops makes it easier to move equipment into or out of the enclosures, particularly by overhead cranes. Ventilating system operates on principle of moving low volume of air at high velocity, and holds to a minimum fumes incident to welding

Five inch flexible tubing, individually counterweighted with light airplane cable, and equipped with screened bell end, can be moved directly over welder's work. High velocity of air picks up fumes from welding operations which would otherwise escape into the air

The second improvement made possible with the new location of the welding booths is concerned with the method of ventilation that has been worked out. Basis of the system is the use of a low volume of air at a fairly high velocity—545 cfm of air per intake at approximately 4 in. static pressure. A 15 ft length of flexible tubing, 5 in. in diameter, and individually counterweighted with light airplane cable, is dropped into each booth from an overhead main ventilating line serving all welding stations.

Each tube has a bell end (screened since one of the welders lost his cap up one of the pipes) and is just the right length to be readily maneuverable around and above the work. No tube is long enough to touch the floor or to be laid on it.

FOR DEFENSE

A new defense construction program was officially launched last month when Ralph A. Bard, assistant secretary of the navy, inspected steam propulsion apparatus under construction at the Westinghouse steam division plant at Lester, Pa., which is working 24 hours a day turning out marine propulsion equipment for fighting ships, cargo vessels and tankers. Production moves in fast—at the right, gears are being hooked to an overhead crane while plant expansion proceeds at the same time

A "lake" on the roof of the building (above) reduces sun heat gain and consequently the load on the air conditioning plant. Although almost half the outside surface of the building is devoted to windows, it is completely sealed to outside air except for entrances and exits. Windows are of glass block construction, but they include a special "island" of double pane clear glass to allow those inside to look out, avoid "claustrophobia"

The recently completed general office for the steam division works is air conditioned, chilled water being piped (through underground cement asbestos piping) from 250 ton steam jet refrigeration equipment in the main boiler house to the air handling equipment in a pent house on the office building roof. There are two similar air handling systems, each supplying four of the eight zones into which the building is divided. The water coils are supplied with heated water in winter. Air is cleaned electrostatically. At the left are some of the controls

Heating · Piping · Air Conditioning

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Effect of Insulation on Plant Performance in the Research Residence

By A. P. Kratz* and S. Konzo**
Urbana, Illinois

SUMMARY—Operating results are reported between an uninsulated and insulated structure utilizing the same heating plant. Data are given for fuel consumption savings due to installation of insulation, and installation of storm sash, and in addition, the reduction in electrical inputs for the stoker and fan motors of the heating plant are tabulated. Temperature data are also given showing the differential between the breathing level and the floor, together with the inside surface temperatures of the exposed walls due to the application of insulation to the structure.

Introduction

THE investigations in gravity warm-air heating and forced warm-air heating in the Research Residence at the University of Illinois prior to 1939 were made in an uninsulated structure. During the summer of 1939 the exposed side walls and ceilings exposed to unheated space were fully insulated with mineral wool.

For the purpose of direct comparison between the results obtained with the uninsulated and insulated structure, the same forced-air heating plant, with no changes in the duct system, the baseboard warm air registers, the centrifugal fan, the stoker-fired warm air furnace, or the automatic control system was used in both cases.

The tests made during the heating season of 1939-1940 were confined to studies of the performance and operating characteristics of the existing heating plant when the heat loss from the structure was materially reduced by the application of insulation and storm sash. Since the existing heating plant was adequate in capacity to heat the uninsulated structure, the use of the same plant in connection with the reduced heat loss was equivalent to the use of an oversized heating plant in the insulated structure.

Description of Research Residence and Heating Equipment

The Research Residence, shown in Fig. 1, and the forced-air heating plant have been completely described in a previous publication.[1]

The Residence is a three-story structure of standard frame construction. The total space heated, including a sun room, amounted to 17,540 cu ft.

In insulating the sidewalls and ceiling, wherever feasible, mineral wool batts were laid between the studding or joists. Wherever batts could not be conveniently used, nodulated mineral wool, having a density of about 5 lb per cubic foot, was blown into place. Owing to the fact that the house was constructed with 6-in. studs, the insulation in the walls and ceilings was 5⅝ in. thick. The insulation in the knee-walls between heated and unheated spaces was 3⅝ in. thick. On account of the presence of ductwork in some of the outside studding spaces, 95 sq ft of exposed wall remained uninsulated as compared with 2,164 sq ft that were insulated. The over-all coefficient of heat transfer for the exposed wall was 0.047 Btu per square foot per hour per degree Fahrenheit and that for the ceilings was 0.048 Btu per square foot per hour per degree Fahrenheit.

The double-hung windows were not equipped with weather strips, and during the tests they remained tightly locked. One series of tests was run with the Residence equipped with storm windows and door. All of the 50 windows on the three stories of the Residence, with the exception of two small quarter-round windows in the east dormitory, were provided with tightly fitting storm sash. Felt stripping was placed along all four contact edges and the storm sash was drawn up tightly to the window casing. The front entrance was equipped with a storm door.

The heating plant consisted of a cast-iron, circular-radiator warm-air furnace used in connection with a forced-air heating system. The principal dimensions of the furnace are shown in Table 1. A centrifugal fan delivered 1675 cu ft of air per minute through the furnace casing and duct system.

The furnace was fired by means of a coal stoker of the underfeed type, which was inserted through the ash pit door. A balanced check damper was installed in the clean out of the chimney. The fuel burned was 1 in. x 10 mesh stoker coal, washed and oil treated and was obtained from Saline County, Ill. The feed rate of the stoker was about 26 lb per hour, and the air input was sufficient to provide a burning rate of approximately 13 lb per hour during *on-periods*.

The heating plant was controlled by means of a room thermostat operating to start and stop the stoker motor, and to start and stop the circulating fan as shown in Fig. 2. This room thermostat was of the heat-anticipating type, and was used in conjunction with two bonnet thermostats which served as high and low limit controls for the temperature of the air in the furnace bonnet. When the room thermostat operated to start the fan and burner, one bonnet thermostat prevented the fan from starting unless the bonnet temperature was above 125 F. The other bonnet thermostat prevented the burner from starting unless the bonnet temperature was below 150 F.

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[1]University of Illinois, *Engineering Experiment Station Bulletin No. 266* by A. P. Kratz and S. Konzo, 1934.
For presentation at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, San Francisco, Calif., June, 1941.

Fig. 1—Warm Air Heating Research Residence, Urbana, Ill.

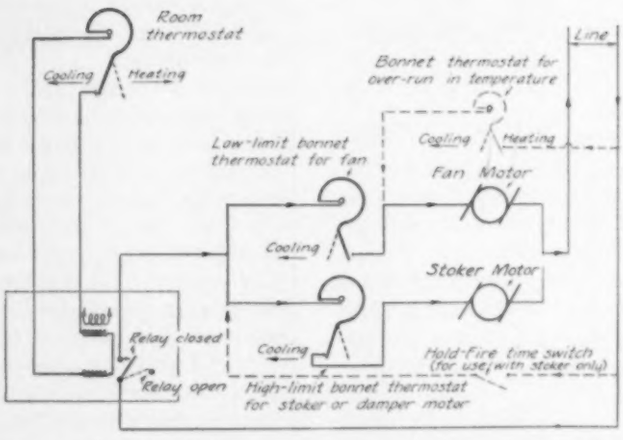

Fig. 2—Simplified wiring diagram for control of fan motor and stoker motor
(Research Residence installation, Seasons of 1937-40)

Table 1—Dimensions and Areas of Furnace Used

Grate diameter	23 in.
Grate area	2.88 sq ft
Firepot diameter	27 in.
Heating surface	
Ash-pit	1.36 sq ft
Firepot	8.10 sq ft
Dome	18.67 sq ft
Radiator	31.93 sq ft
Total	60.26 sq ft
Ratio of heating surface to grate area	20.9
Casing diameter	50 in.
Free area through casing	4.97 sq ft
Combustion space[a]	8.8 cu ft
Free area through over-fire damper	4.9 sq in.

[a] Combustion space is defined in this case as the space above the hearth level, including the dome, but not the feed neck.

Method of Conducting Tests

The average of the air temperatures in all of the rooms of the house was maintained at 72 F both day and night. Observations of weather, indoor room air temperatures, room relative humidities, and other incidental data were made daily at 7:00 a.m., 11:00 a.m., 4:00 p.m., and 10:00 p.m. Complete data were obtained for each 24-hour test period on the fuel consumption, weight of ash and clinkers removed, the total integrated time of operation of the fan and of the stoker, the total electrical input to the fan and burner motors, and the total number of *on-periods* of both the circulating fan and the stoker. Daily observations were made of the volume of air circulated, and the filters were cleaned with sufficient frequency to maintain the air volume constant. In addition, continuous records of temperatures, CO_2, and the index of smoke density were obtained for each 24-hour period. Each day at 11 a.m., the clinkers were removed, the fuel bed was levelled, and the hopper was filled with coal. During extremely mild weather no attention was given the fuel bed or hopper, except as required every two or three days. By means of the balanced check damper, the draft in the smoke pipe was maintained at approximately 0.05 in. of water. For each series of tests, data were obtained over a wide range of outdoor weather conditions.

The following series of tests were run: *Series 4-38*, which was discussed in a previous paper,[2] and for which the Residence was not insulated and the windows were not equipped with storm sash; *Series 1-39*, for which the Residence was insulated, but the windows were not equipped with storm sash; and *Series 2-39*, for which the Residence was insulated and the windows were equipped with storm sash. In no case were any changes made in the quantity of air circulated, in the feed rate and the rate of air supply to the stoker, in the settings of the automatic controls, or in the method of distribution of air from the baseboard registers.

Results of Tests

Calculated Values of Heat Loss: The calculated values of the heat losses from the Residence, based on coefficients computed from the conductivities given in THE ASHVE GUIDE and on a temperature difference between indoors and outdoors of 80 F, are given in Fig. 3. The heat losses for each of the three stories of the Residence, as well as those for the entire structure, are shown in graphical form. As an example, in *Case A*, for which the

[2] Performance of a Stoker-Fired Warm-Air Furnace as Affected by Burning Rate and Feed Rate, by A. P. Kratz and S. Konzo. ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*, January 1940, p. 55.)

house was not insulated and storm sash were not used, the calculated heat losses were 32,385 Btu per hour for the third story, 49,780 for the second story, 67,675 for the first story, and 149,840 for the entire structure. In the usual two story structure of compact design and without exposed wings, the heat loss from the second story rooms is generally greater than that from the first story rooms. In the case of the Research Residence, however, the exposed sun room added a material heat loss to that from the remainder of the rooms on the first story, with the result that the total heat loss on the first story was much greater than that on the upper stories.

In *Case C*, for which the house was insulated and storm sash were not used, the calculated heat losses were 13,255 Btu per hour for the third story, 28,167 for the second story, 50,551 for the first story, and 91,973 for the entire structure. The ratios of each of these values to the corresponding values obtained for *Case A* are given in the last column in Fig. 3, and were 0.409, 0.562, 0.746, and 0.614 for the third story, second story, first story and entire structure respectively. That is, the estimated percentage reduction in the heat loss effected by the application of insulation to the sidewalls and ceiling was considerably greater on the third and second stories than it was on the first story. Furthermore, while heat losses from the second story of the more common type of two story structure of compact design are greater than those from the first story, usually the application of insulation to sidewalls and ceiling also results in a proportionally greater reduction on the second story than on the first story.

After the Residence had been insulated, and before any changes had been made in the settings of the dampers in the branch ducts leading to the twelve warm air registers, a few preliminary tests were made to determine whether the installation of the insulation had resulted in any material lack of uniformity in the room temperatures. From the reductions in heat loss shown in *Case C* in Fig. 3, it would be reasonable to expect that after the application of insulation, the room temperatures in the third and second story rooms would be higher

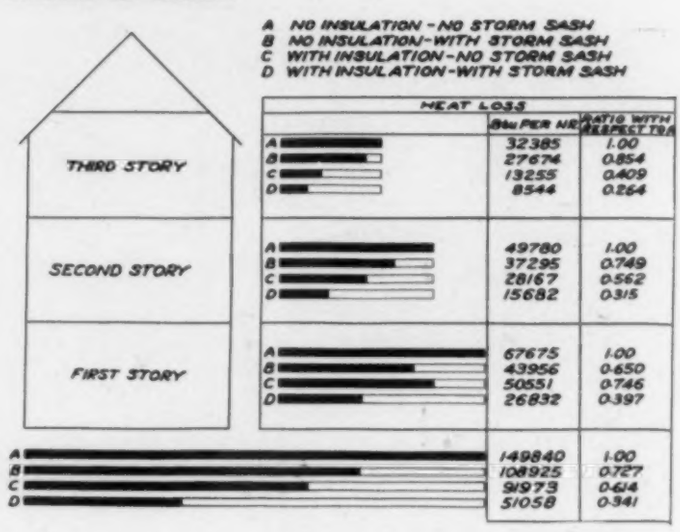

Fig. 3—Calculated heat losses from Research Residence for 80 F difference

than those in the first story rooms. Measurements of the air temperatures in the various rooms of the Residence confirmed this hypothesis by proving that the temperatures in the third story rooms were from 5 to 7 F higher, and the temperatures in the second story rooms were from 1 to 3 F higher, than those in the first story rooms.

Hence, in the case of many existing homes in which the heating facilities in upper story rooms are inadequate, the application of insulation to sidewalls and ceiling should not only reduce the total heat losses, but should also tend to bring about a better balance in temperatures in the rooms on the two stories.

A comparison of *Case B* in Fig. 3 with *Case A* indicates that the application of storm sash alone on an uninsulated house is relatively more effective in reducing the heat losses from the first story rooms than it is on the upper stories. Whereas, a comparison of *Case D* with *Case A* indicates that the combined application of insulation and storm sash would reduce the heat losses approximately in proportion, on each of the three stories. In any case, after the application of storm sash or insulation to an existing structure, some amount of readjustment of the heat inputs to the various rooms will be necessary.

After the preliminary observations were made, the dampers in the ducts of the forced-air heating system in the Research Residence were readjusted to reduce the amount of warm air delivered to the second and third story rooms. The damper adjustments were easily made and no difficulty was experienced in obtaining a uniform air temperature of about 72 F in all of the rooms in the Residence.

Fuel Consumption: The calculated values of the total heat loss for *Cases A* and *C*, given in Fig. 3, indicate an estimated possible reduction in heat loss effected by insulating the sidewalls and ceiling amounting to 38.6 per cent. The data presented in Fig. 4, showing the weight of equivalent coal burned per day, were obtained from actual tests and afforded a means of comparing the actual reduction in fuel consumption with the calculated reduction in heat loss. The average reduction in fuel consumption obtained over a wide range of weather conditions was about 30 per cent, or approximately 0.78 of the estimated reduction in heat loss. The agreement between the actual reduction in fuel consumption and the estimated reduction of heat loss may be considered as close as can be expected considering the uncertainties in many of the assumptions inherent in the calculation of heat losses.

A similar comparison from Fig. 3 of the calculated heat losses for *Case A* and *Case D*, for the latter of which both insulation and storm sash were used, indicates an estimated possible total reduction of 65.9 per cent, or an additional esti-

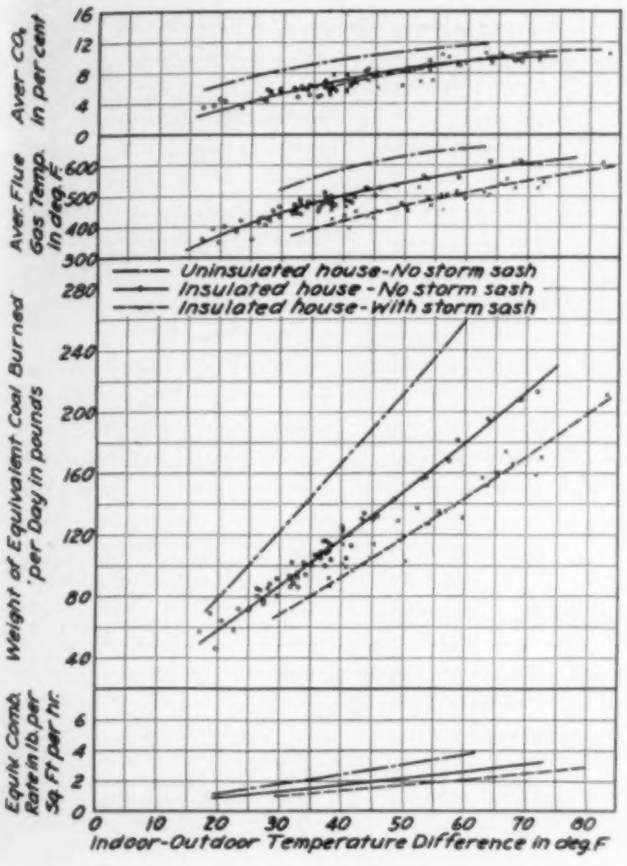
Aver CO2 in per cent
Aver. Flue Gas Temp. in deg. F.
Weight of Equivalent Coal Burned per Day in pounds
Equiv Comb. Rate in lb. per Sq Ft per hr
Uninsulated house-No storm sash
Insulated house - No storm sash
Insulated house-With storm sash
Indoor-Outdoor Temperature Difference in deg.F

Fig. 4—Furnace performance

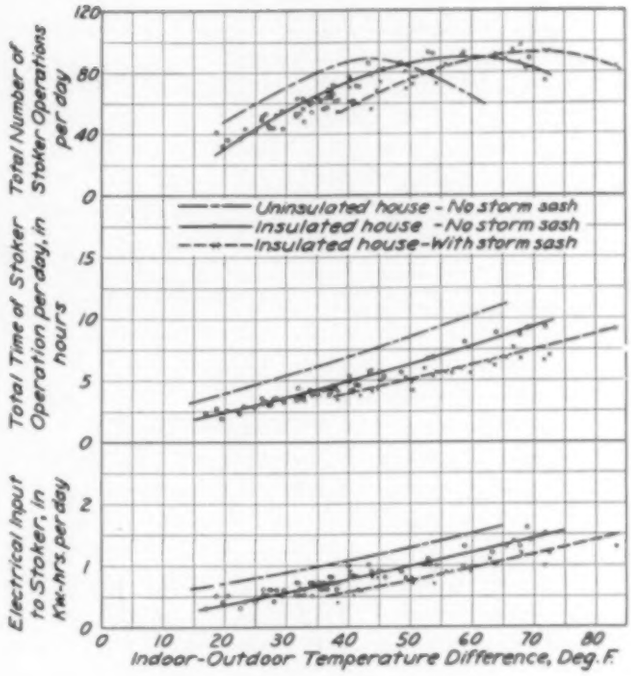
Total Number of Stoker Operations per day
Total Time of Stoker Operation per day, in hours
Electrical Input to Stoker, in Kw.-hrs. per day
Uninsulated house - No storm sash
Insulated house - No storm sash
Insulated house-With storm sash
Indoor-Outdoor Temperature Difference, Deg. F.

Fig. 5—Operating characteristics of stoker

mated reduction of 27.3 per cent effected by the storm sash alone. The actual reduction in fuel consumption resulting from the use of both insulation and storm sash as shown in Fig. 4 was about 45 per cent, or only 0.68 of the estimated possible reduction in heat loss. The reduction in fuel consumption resulting from the use of storm sash alone was about 15 per cent or only 0.55 of the estimated reduction in heat loss. In this case, the discrepancy between the estimated reduction in heat loss and the actual reduction in fuel consumption was quite large and may be due to the fact that certain factors affected by the actual conditions of operation deviated from those assumed for the calculation of the heat loss.

Part of the discrepancy may be attributed to the uncertainties which necessarily accompany the computation of infiltration losses, a complete discussion of which has been presented in a previous paper.[3] Some of the discrepancy may have resulted from the use of an oversized stoker-fired furnace in the insulated Residence. The stoker and furnace, which were adequate in size for the uninsulated house, were operated intermittently, by means of the thermostatic control system, to maintain the desired room temperatures in the insulated house. For a given outdoor temperature, the total operating time of the stoker was not as long, as shown in Fig. 5, and the average CO_2 in the flue gas was not as high, as shown in the top of Fig. 4, in the case of the insulated house as they were in the case of the uninsulated house. It is possible that in spite of the decrease in the accompanying average flue gas temperatures, as shown in Fig. 4, the over-all combustion efficiency for the stoker, during both the *on-periods* and *off-periods*, was less with the stoker and furnace actually used than it would have been if the equipment had been properly sized for the insulated house. No tests have as yet been made to determine whether the use of a smaller sized stoker and furnace in the insulated Residence would improve the over-all combustion efficiency and thereby reduce the discrepancy between the estimated reduction in heat loss and the actual reduction in fuel consumption. However, previous tests[4] would indicate that if the stoker used had been regulated in the case of the insulated house to burn coal at a rate less than the 13 lb per hour actually employed, some gain in fuel economy would have been attained. In general, if maximum fuel economies are to be obtained after the heat losses from a given house have been reduced by insulation or otherwise, some attention should be given to the matter of insuring the use of a burning rate commensurate with the reduced heat losses. On the whole the results obtained are representative of those arising from the usual practice of insulating the house without changing the heating plant.

Furnace Capacity: As shown by Fig. 4, the average flue gas temperatures for a given day were greatest when the house was not insulated and storm sash were not used, and were the least when the house was insulated and equipped with storm sash. Hence, the application of insulation and storm sash to a house already equipped with a furnace plant is equivalent to lightening the maximum load on the

[3]Fuel Saving Resulting from the Use of Storm Windows and Doors, by A. P. Kratz and S. Konzo. (ASHVE TRANSACTIONS, Vol. 42, 1936, p. 87.)

[4]Loc. Cit. Note 2.

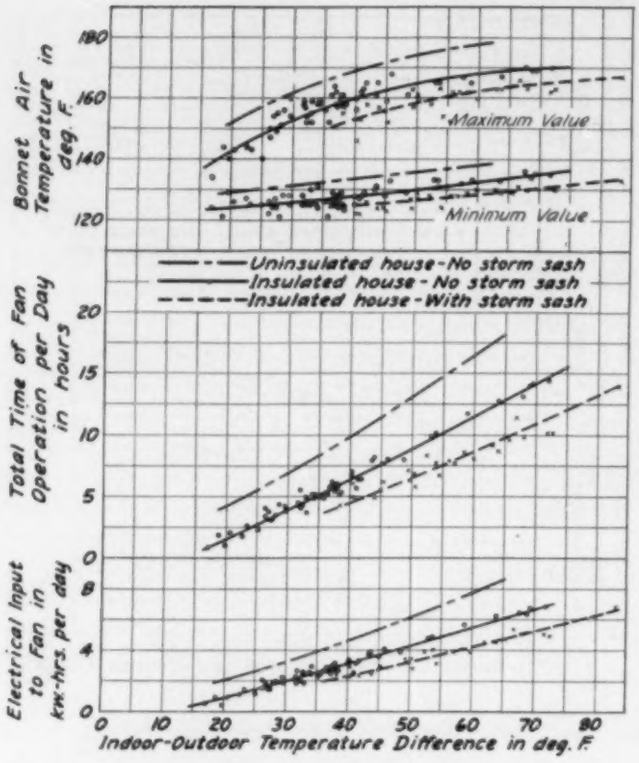
Bonnet Air Temperature in deg. F.
Maximum Value
Minimum Value
Uninsulated house - No storm sash
Insulated house - No storm sash
Insulated house - With storm sash
Total Time of Fan Operation per Day in hours
Electrical Input to Fan in kw-hrs. per day
Indoor-Outdoor Temperature Difference in deg. F.

Fig. 6—Operating characteristics of fan

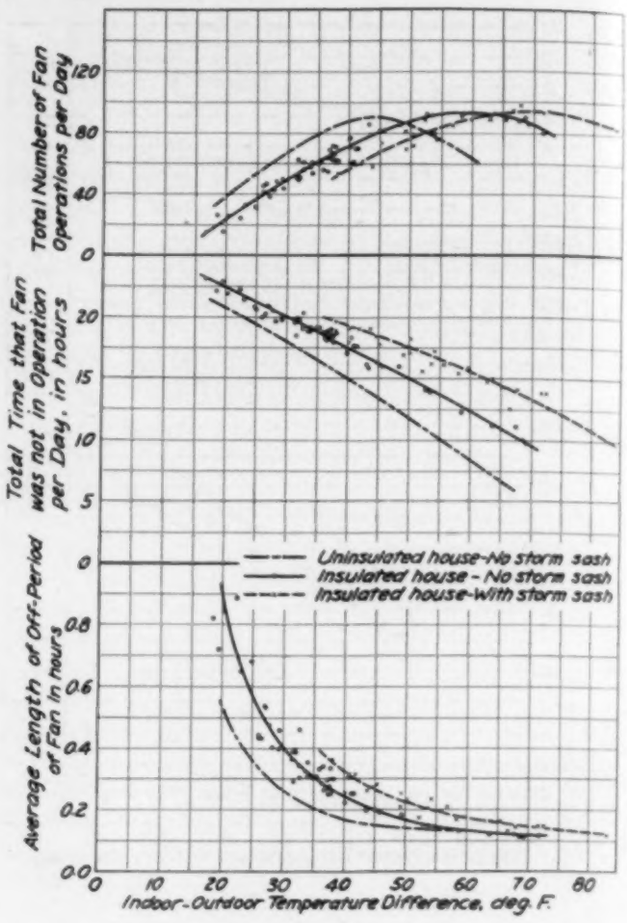
Total Number of Fan Operations per Day
Total Time that Fan was not in Operation per Day, in hours
Average Length of Off-Period of Fan in hours
Uninsulated house-No storm sash
Insulated house - No storm sash
Insulated house-With storm sash
Indoor-Outdoor Temperature Difference, deg. F.

Fig. 7—Off-periods of fan

plant, and thereby increasing the life of the furnace equipment.

The furnace and stoker in the Residence had sufficient capacity to satisfy the heating demands of the uninsulated house when the outdoor temperature attained a value of —10 F, corresponding to the design temperature used in Urbana, Ill. By burning the same total amount of coal as that required to heat the uninsulated house at —10 F, the furnace and stoker in the insulated house would have had adequate capacity to satisfy the heating demands for an outdoor temperature of approximately —44 F, which is far in excess of the lowest temperature recorded in Urbana.

In an old house this excess capacity represents a reserve sufficient to provide for the lowest possible extreme of outdoor temperatures in a given locality, while in a new house it represents the margin by which the initial cost of a new plant may be reduced.

Stoker and Fan Operation: The influence of the controls on the fan and stoker motors was such that, irrespective of the size of the furnace relative to the over-all heating demand of the house, the heat output was proportioned approximately to the heat losses occurring in either the insulated or the uninsulated house at a given indoor-outdoor temperature difference. In the case of the insulated, as compared with the uninsulated house, since the heat demand was less, the stoker remained in operation for a shorter period. Hence, at the end of this period the combustion was not as active and the amount of incandescent coke in the fuel bed was less. As a result, the combustion during the *off-period* was also less active, and the temperature in the bonnet at the end of the *off-period* and hence at the start of the subsequent *on-period* was not as high. During the *on-period,* while air was being circulated by the fan, the temperature in the bonnet decreased, and at the end of the *on-period* it was lower than that obtained in the case of the uninsulated house. Consequently, as shown in Fig. 6, the furnace operated at a lower range of bonnet temperatures in the case of the insulated house than it did in the case of the uninsulated house. This lower temperature range, in conjunction with the reduced time of operation, served to proportion the furnace output to the heat loss from the house. As shown by Figs. 5 and 6, the total time of operation of the stoker and fan per day, and the electrical inputs to the stoker motor and fan motor per day were each decreased approximately 25 to 30 per cent after the house was insulated, as compared with the results obtained in the uninsulated house.

Factors Affecting Comfort: As discussed in a previous publication,[5] the maintenance of a small temperature differential from breathing level to floor is a necessary requirement for comfort. Furthermore, short *off-periods* of the fan are more desirable from the standpoint of comfort than long *off-periods,* in that the shorter *off-periods* tend to

[5]Study of Methods of Control and Types of Registers as Affecting Temperature Variations in the Research Residence, by A. P. Kratz and S. Konzo. (ASHVE Transactions, Vol. 44, 1938, p. 309.)

Investigation of Oil-Fired Forced-Air Furnace Systems in the Research Residence, by A. P. Kratz and S. Konzo. (University of Illinois, *Engineering Experiment Station Bulletin* 318, Nov. 1939.)

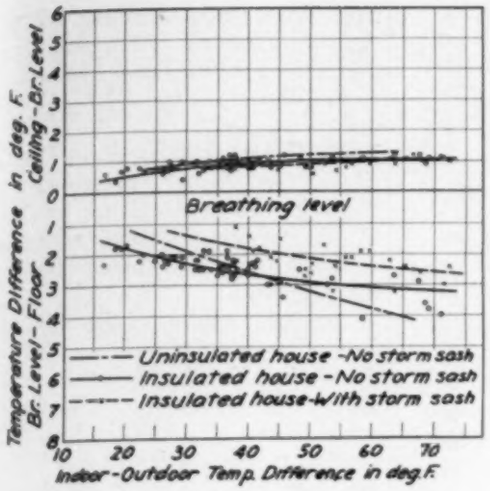
Temperature Difference in deg. F.
Br. Level - Floor
Ceiling - Br. Level
Breathing level
Uninsulated house - No storm sash
Insulated house - No storm sash
Insulated house - With storm sash
Indoor - Outdoor Temp. Difference in deg. F.

Fig. 8—Room temperature differentials

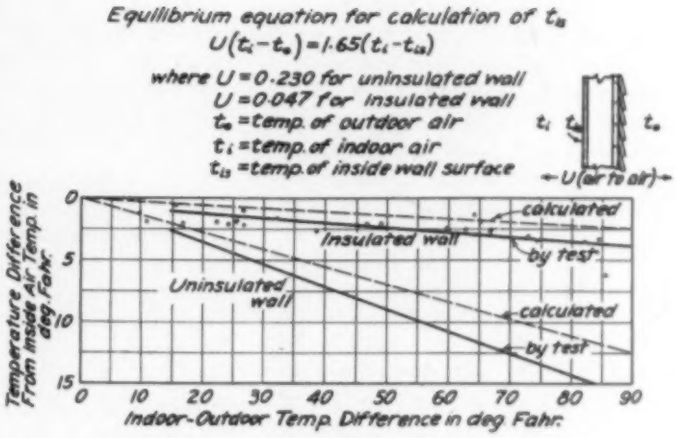
Equilibrium equation for calculation of t_{is}
$U(t_i - t_o) = 1.65(t_i - t_{is})$
where U = 0.230 for uninsulated wall
U = 0.047 for insulated wall
t_o = temp. of outdoor air
t_i = temp. of indoor air
t_{is} = temp. of inside wall surface
U (air to air)
Temperature Difference From Inside Air Temp. in deg. Fahr.
Insulated wall
Uninsulated wall
calculated
by test
calculated
by test
Indoor - Outdoor Temp. Difference in deg. Fahr.

Fig. 9—Inside surface temperatures of wall (North wall, living room)

minimize the temperature variations in the room brought about by cyclic operation. This is particularly true in cold weather, or for temperature differences from indoor to outdoor exceeding approximately 50 F.

The tests made in the uninsulated and insulated Residence afforded a means of comparing the length of *off-periods,* and also the temperature differentials as affected by changes in the nature of the structure when no changes were made in the heating plant itself, or in its method of operation. As shown in Fig. 7, for a given temperature difference from indoors to outdoors, the use of insulation brought about an increase in the length of time that the fan was not in operation per day. Fortunately, for temperature differences exceeding about 50 F, an increase in the number of fan operations per day was also obtained, with the net result that the average length of the *off-periods* was only slightly increased.

During periods of cold weather, as shown by Fig. 8, the temperature differentials from breathing level and floor were diminished slightly after the house was insulated, probably as a result of the fact that any down currents of air over the exposed walls were not as cold in the case of the insulated as they were in that of the uninsulated house. Hence, any slight tendency for the existence of a cold floor resulting from the increase in the length of the *off-period* of the fan evidently was more than offset by the beneficial effect of the warmer air coming down the exposed walls, and the temperature differential from breathing level and floor was decreased rather than increased. On the whole, no marked differences in comfort, as measured by room air temperatures alone, could be attributed to the insulation.

In this connection, the data given in Fig. 8 show that the combined use of insulation and storm sash brought about a reduction of more than 1 F in the temperature differential from breathing level to floor. For example, for an indoor-outdoor temperature difference of 60 F, the use of insulation and storm sash reduced the temperature differential from breathing level to floor from a value of 3.8 F to one of 2.3 F. The temperature differentials shown in Fig. 8 were the averages of readings made in eleven different locations in the Residence, and at most were much smaller than those that have been obtained with other arrangements of the heating plant in the same structure. It is possible that in structures which are not as satisfactorily heated as the Residence originally was, and which show much larger temperature differentials from breathing level to floor, the use of insulation and storm sash might show to much better advantage.

The most marked improvement in comfort with the use of insulation was that resulting from the large increase in the inside surface temperatures of the exposed walls. The difference in temperature between the indoor air and the inside wall surface was plotted, as shown in Fig. 9, against the temperature difference between the indoor air and outdoor air. The temperature differences between indoor air and inside wall surface obtained from test observations on both the uninsulated and insulated wall section are designated by solid lines, and calculated values of the same quantities are designated by broken lines. For both the uninsulated and insulated wall sections, the calculated temperature differences were slightly smaller than the observed differences. The calculated and observed differences are in reasonably good agreement considering the fact that the actual coefficients of heat transmission for the wall and the surface coefficient may differ somewhat from those assigned in the calculations.

Test values in Fig. 9 indicate that for an indoor-outdoor temperature difference of 80 F the wall surface temperature was 14.5 F lower than that of the indoor air in the case of the uninsulated wall and only 3.5 F lower in the case of the insulated wall, or an improvement of 11.0 F effected by the insulation. With the large wall areas involved, this warmer wall surface temperature would prove to be a significant factor in reducing the radiation loss from the body, and in increasing the comfort of the occupant, even though the room air temperature was the same. In this connection it is of interest to note that the increase in inside surface temperature of the glass in the windows brought about by the use of storm sash is of even greater magnitude than that shown in Fig. 9 for the

wall. The reductions in fuel consumption, shown in Fig. 4, effected by the use of insulation and storm sash would probably have been greater if the tests had been made so that in both cases equal comfort had been maintained in the Residence, rather than equal dry-bulb temperatures. Due to difficulties in evaluating comfort, and to the fact that wall surface temperatures were not measured in all of the rooms of the Residence, no such tests were attempted. In any case the fuel economies reported in this paper may be considered as minimum rather than maximum values.

Summary

The following is a summary of the results obtained in the Research Residence and is applicable to the conditions under which the tests were conducted:

1. The installation of insulation resulted in an average saving of approximately 30 per cent in the actual fuel consumption, as compared with an estimated reduction in heat loss of 38.6 per cent.

2. The installation of storm sash on the insulated house resulted in an additional average saving of approximately 15 per cent in the actual fuel consumption as compared with an estimated reduction of 27.3 per cent.

3. The electrical inputs to the stoker and fan motors per day were each decreased from 25 to 30 per cent after the house was insulated, as compared with the results obtained in the uninsulated house.

4. The estimated reductions in heat losses indicated that a greater benefit was obtained from the insulation of the second and third stories than from that of the first. This was confirmed by the fact that temperatures on the second story were from 1 to 3 F higher, and those on the third story were from 5 to 7 F higher, after the installation of insulation and before final adjustment of the dampers, than they were in the uninsulated house.

5. The temperature differential from breathing level to floor was only slightly decreased by the application of insulation.

6. At an indoor-outdoor temperature difference of 80 F, the temperature of the inside surface of the exposed walls was increased approximately 11.0 F by the application of insulation, thus resulting in a material potential increase in comfort.

7. The application of insulation without changes in the heating system was equivalent to the use of an oversized furnace. In spite of this, on the whole no difficulties were encountered from overheating or from lack of uniform temperature distribution in the rooms and the results may be considered as representative of those arising from the usual practice of insulating the house without changing the heating plant.

Acknowledgments

The results presented in this paper were obtained in connection with the investigation of warm-air furnaces and heating systems in the Research Residence[6] at the University of Illinois, conducted by the Engineering Experiment Station of which M. L. Enger, Dean of the College of Engineering, is the director, and in the Department of Mechanical Engineering of which O. A. Leutwiler, Professor of Mechanical Engineering Design, is the head. This investigation is a cooperative project sponsored jointly by the *National Warm Air Heating and Air Conditioning Association* and the Engineering Experiment Station. These results will ultimately comprise part of a bulletin of the Engineering Experiment Station. Acknowledgment is made to D. W. Thomson, formerly Research Graduate Assistant, for his services in conducting the tests and in the reduction of the test data. Acknowledgment is also made to the *National Mineral Wool Association* for their interest and cooperation in insulating the Residence, and to the various manufacturers who cooperated by furnishing instruments and equipment.

[6]The Research Residence in Urbana, Illinois, was built, furnished, and completely equipped specifically for research work in warm-air heating by the *National Warm-Air Heating and Air Conditioning Association* in December, 1924.

PITTSBURGH HEARS TALK ON TEMPERATURE CONTROL

March 10, 1941. Pres. E. C. Smyers called the March meeting to order at 8:00 p.m. at the Hotel Roosevelt, and the minutes of the previous meeting were approved as read.

F. B. Mahon announced for the program committee that Prof. E. O. Eastwood, First Vice-President of the Society, would visit the Pittsburgh Chapter in April.

In accordance with the recent amendments to the By-Laws, President Smyers appointed the following to serve on the nominating committee: A. F. Nass, *Chairman*; J. F. Collins, Jr., and R. J. J. Tennant.

Mr. Collins, member of the Council of the Society, presented P. Nicholls, Fuels Engineer at the U. S. Bureau of Mines, with his certificate of Life Membership in the ASHVE.

The guest speaker of the evening, J. R. Vernon, Chicago, was introduced by F. C. McIntosh, and spoke on automatic temperature control for air conditioning and heating systems in larger buildings. He presented this as a completely illustrated lecture on all types of thermostats, humidistats, etc., and concluded his remarks by showing the various devices assembled into several types of complete systems of temperature control. The audience thoroughly enjoyed his talk, and, upon motion of P. A. Edwards, Mr. Vernon was extended a rising vote of thanks.

According to Secretary Rockwell's report 60 members and guests were in attendance at this meeting which adjourned at 10:15 p.m.

PHILADELPHIA CHAPTER IS PLANNING FOR YOU in '42 . . . . .

. . . . . 48th ANNUAL MEETING OF ASHVE and the 7th International Heating and Ventilating Exposition

January 26-30, 1942

Effect of Radiation Upon Thermal Conductivities Determined by Hot Plates

By R. M. Johnston* and Carl B. Ruehr**, Blacksburg, Va.

THE accepted method of determining the thermal conductivities of homogeneous insulating materials is by use of the guarded hot plate as specified by the Society for this purpose. At present there are two general types of hot plates; those of metallic construction having copper or aluminum surfaces, and those made of refractory material, notably the alundum plate developed and described in previously available literature by Heilman.[1] Both types employ one or more guard rings, and the methods of operation are similar. However, thermal conductivities determined by the two plates may show a consistent disagreement even when determined by the same investigator using exactly the same apparatus with the exception of the hot plate. This persistent variation was noted by the authors and investigations were conducted, the purpose of which was to determinate and eliminate the cause for disagreement, or the development of a method of compensating for it, the results of which are presented in this paper.

The apparatus used in this work was furnished by the Engineering Experiment Station of the Virginia Polytechnic Institute and the work was conducted as a part of the research program of this institution. The hot plate was of the alundum type and was made by the Mellon Institute, and is approved by the Society and by the *National Heat Transmission Council*.

The cold plates were designed by the Engineering Experiment Station and consist of cast brass cups ½ in. deep and 8½ in. inside diameter which are covered by 3/16-in. copper plates attached by ring welds. Holes are drilled in opposite sides of the cup for the admission of cooling water. The upper surface is tapered and filleted to a finished surface 3⅝ in. in diameter and ½ in. thick which contains four threaded holes by which floor flanges may be bolted to the plates to facilitate their attachment to the walls of the test chamber, as illustrated in Fig. 1. The copper face plates contain four milled slots in which constantan wires are run from their contact points with the plate 29/32 in., 1⅛ in., 2½ in. and 3¼ in., respectively, from the centers of the plates; and these plates constitute the common wire for all four thermocouples. The constantan wires are soldered to the copper plates at their contact points and are insulated from the plates at all other points.

Direct current was used which was supplied by a motor-generator set, and the power input to the plates was determined by the use of a volt-meter and an ammeter. Although not precision instruments, both were calibrated and the power consumed was calculated from corrected readings as determined by calibration. The potentiometer used was an instrument having an accuracy of 1/100 millivolts.

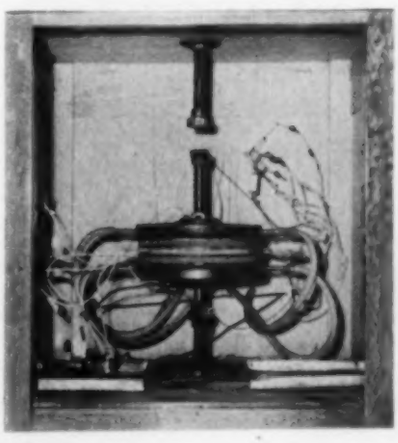

Fig. 1—Assembly of plates and samples in test box

The electric circuits to the central heating element and to the guard rings of the hot plate were furnished with suitable slide wire rheostats such that accurate balancing was obtained. These were used for final adjustments only, the main voltage regulation being accomplished at the motor-generator control board.

A metallic surface hot plate was not available. However, a simple arrangement was devised to give the same characteristic performance. Two 9-in. diameter hot rolled copper plates 1/16 in. thick were placed on each side of the alundum plate. To insure true plane surfaces these plates were subjected to a pressure of 30,000 lb per square inch in a hydraulic press and the slight irregularities remaining after pressing were eliminated by machining the surfaces. This arrangement gave, in effect, a copper surfaced hot plate, the alundum plate comprising the heating elements.

Two additional copper plates, similarly surfaced, were provided for use between the samples and the cold plates. These were employed so that plate characteristics on both sides of the sample could be controlled.

In making the tests, these plates were assembled in the test box as shown in Fig. 1. All cracks in the test box were puttied and the removable panel was bolted in place to minimize air leakage, and moisture effects, during the test runs. As a further assurance that no moisture regain would take place in the samples, pans of calcium chloride

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**Graduate Fellow in Mechanical Engrg., Virginia Polytechnic Institute.

[1] Apparatus for the Determination of Thermal Conductivities of High Temperature Insulation, by R. H. Heilman (*American Institute of Chemical Engineers*, June, 1936).

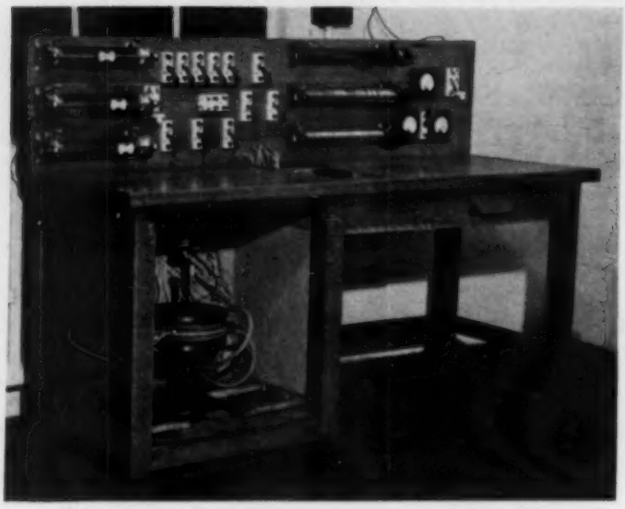

Fig. 2—Test apparatus, test box panel removed

were placed in the test box prior to the test and remained in the box for the duration of all test runs.

Two 9-in. diameter samples were cut from a piece of ½ in. rigid fiber board insulation. These were used for all tests, and were maintained at 215 F in an electric drier for a period of 24 hours prior to each test run.

It was assumed that the variations in conductivity values obtained from different hot plates were due to variations in the surface characteristics of the plates. For this reason the emissivities of the copper surfaces facing the samples were varied by painting with flat black lacquer, white lacquer, 10 per cent aluminum paint, and finally by polishing the plates to a state of high reflectivity. The emissivities obtained in this manner were 0.96, 0.80, 0.52, and 0.03, respectively.[2] The emissivity of the surfaces of the sample was taken as 0.93 and of the alundum plate as 0.91.

The thermocouples used for the determination of the surface temperatures of the samples were No. 26 B and S gage copper-constantan wire with enamel insulation. Four thermocouples were used with each sample, two on the hot side and two on the cold side. These were sewed to the surfaces of the samples with a very fine thread, one being placed at the center of the sample and another at a position corresponding to the inner guard ring of the hot plate. The thermocouples built into the alundum plate and those soldered to the cooling plates were not used when tests employing the copper insert plates were made, except in the preliminary balancing.

Briefly, the theory involved in the use of the hot plate method is that a measurable quantity of heat, developed by an electric current in a central section of the plate, is constrained by secondary heating elements so that its path of flow is normal to the surface of the plate and thus, normal to the surfaces of the sample which are in contact with or face the plate. All of the heat developed in the central or primary heating element and a part of that developed by the secondary elements flows through the sample and is finally absorbed by cold plates through which a cooling medium is circulated. Thus, with the thickness, test area and surface temperatures of the sample known, the thermal conductivity of the samples may be easily calculated using the equation,

$$k = \frac{3.415(EI)L}{A\,(t_1 - t_2)}$$, in which

E = the voltage drop across the primary element

I = the current flowing through the primary element (amperes)

L = the thickness of the sample (inches)

A = the area of the test section (square (inches)

t_1 = the temperature on the hot side of the sample (degree Fahrenheit)

t_2 = the temperature on the cold side of the sample (degree Fahrenheit)

The effective emissivity of the plate and sample was calculated on the basis of the equation[3]

$$e_{eff} = \frac{1}{(1/e_p) + (1/e_s) - 1}$$, in which

e_{eff} = the effective emissivity of the facing surfaces

e_p = the emissivity of the plate

e_s = the emissivity of the sample.

In making the tests, an equalization period of approximately 2½ hours was required for the establishment of thermal equilibrium in all cases: that is, to establish a constant temperature condition throughout the test and inner guard ring area. When this condition existed, thermocouple and power readings were taken at intervals of 10 min for 1 hour, these being averaged for the conductivity calculations.

A total of 14 tests was made. Of these, tests Nos. 3, 4, 5, 6, 7, 8, 9, 10, 13, and 14 were made with plate inserts on both sides of the sample; Nos. 1 and 2 were made without plate inserts; test No. 11 was made with black lacquered plates on the hot sides only; and test No. 12 was made with highly polished plates on the hot sides only. Data and results of these tests are included in Table 1. Fig. 3 shows graphically the results of test Nos. 1 through 10; and the results of tests Nos. 10, 13 and 14 are plotted in Fig. 4.

Since the observed thermal conductivity is known to be affected by the mean temperature at which the tests are made, tests Nos. 13 and 14 were made so that the variation of thermal conductivity with mean temperature might be determined for the samples used. These tests were made with the same plate inserts having the same emissivities as were used for tests Nos. 9 and 10, the results being plotted with the results of test No. 10 as previously stated. This influence was anticipated and the mean temperature of tests Nos. 1 through 10 was maintained within a maximum spread of 8.1 F.

The effect of mean temperature was determined for one emissivity only. However, this variation is a characteristic property of the material and thus should not be affected in any way by the test apparatus, regardless of any other results which may be attributed to it. That is, variations of mean test temperature made with black lacquered plates should give results which when plotted in Fig. 4 would produce a straight line having the same slope as the line shown but having a much greater value at the ordinate intercept. Therefore, Fig. 4 may be used to correct the points of the emissivity-conductivity curve for mean temperature variation, if desired. It may be pointed out, however, that the maximum correc-

[2] Heat Transmission, by W. H. McAdams (McGraw-Hill Co.).

[3] Loc. cit. Note 2.

Table 1—Data and Results of Tests

Test No.	Effective Emissivity	Temperature Warm Side Deg F	Temperature Cold Side Deg F	Mean Temperature Deg F	Corrected Voltage	Corrected Amperage	Power Input Watts	Calculated Conductivity
1	0.85	120.0	52.0	86.0	9.3	0.58	5.39	0.345
2	0.85	128.5	56.4	92.5	9.5	0.60	5.40	0.344
3	0.89	118.8	57.2	88.0	9.4	0.58	5.45	0.385
4	0.89	120.5	57.2	88.8	9.5	0.59	5.61	0.387
5	0.75	128.8	56.4	92.6	9.0	0.57	5.14	0.307
6	0.75	127.2	61.0	94.1	8.6	0.54	4.64	0.305
7	0.50	128.0	56.8	92.4	8.4	0.54	4.54	0.277
8	0.50	126.8	56.8	91.8	8.4	0.53	4.46	0.276
9	0.03	124.8	56.8	90.8	7.6	0.49	3.72	0.238
10	0.03	126.8	56.8	91.8	7.7	0.50	3.85	0.239
11	0.89	115.2	54.0	80.6	9.3	0.58	.....	0.383
12	0.03	114.4	57.6	81.0	7.7	0.50	.....	0.294
13	0.03	88.8	56.4	72.6	5.1	0.34	.....	0.233
14	0.03	169.2	60.0	114.6	9.9	0.62	.....	0.246

tion thus obtained is only 0.00259 Btu per square foot per hour per degree Fahrenheit. On this basis, such a correction is of little value and may be neglected without introducing noticeable error.

From the tabulated results and from Fig. 3 it is evident that an increase in the emissivity of the plates causes an increase in the observed values of the thermal conductivity of the samples used. This variation may be caused by, (1) contact resistance, (2) penetration of the surface of the sample by radiant heat, the effects of both being aggravated by the surface condition of the material tested, and (3) the relative emissivities of the thermocouples and the sample surface.

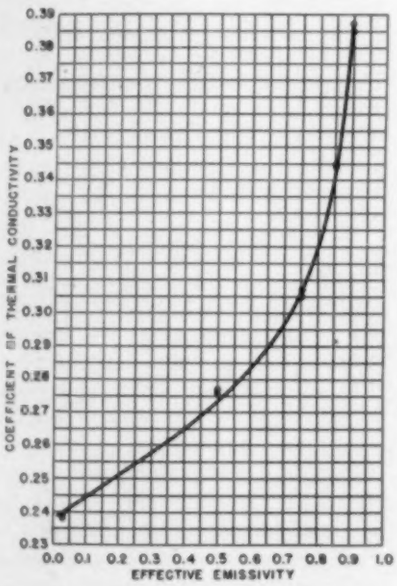

Fig. 3—Variation of coefficient of thermal conductivity with emissivity of plate

Thus, since the material is fibrous and the surfaces fuzzy and discontinuous due to slight irregularities the presence of air in the surfaces of the sample must be recognized. The air thus trapped between the plates and the surfaces of the samples forms a film which, despite its discontinuity and thinness, resists the flow of heat to the sample. Furthermore, a part of the heat flow through those parts of the surfaces thus filmed will be radiant heat. In this case, thermocouples placed between the surfaces of the sample and the hot or cold plate will not indicate a true surface temperature, thermocouple readings thus obtained giving either the average of the plate and sample surface temperatures or the mean temperature of the air film if its thickness is such that the thermocouple is not in contact with either surface. Due to the weight of the plates and their arrangement during tests, the authors believe that the thermocouples employed were in contact with both the surface of the sample and the surface of the adjoining plates in all cases, and that where a thermocouple was within a filmed area it indicated the average of the two surface temperatures. Under these circumstances the surface temperature of a plate having a high emissivity will be lower than the surface temperature of the same plate when transmitting the same amount of heat but having a low emissivity. Thus, with the cold plate temperature held constant, the measured temperature drop across the sample is lower for a highly emissive surface than for one of low emissivity; and the calculated thermal conductivity is therefore greater for the highly emissive surface.

With regard to the penetration of the surface of the sample by radiant heat, it is suggested that the surface porosity and large fiber size constitute a surface structure analogous with those of a low density nature as reported by Finck,[4] who advances the opinion that for low density fibrous materials the effect of radiation upon thermal conductivity values is of much greater importance than is generally realized. Although not primarily interested in radiation, Finck reports a change in thermal conductivity of 17 per cent obtained by changing the emissivity characteristics of a fibrous material by dusting it with aluminum powder. Since the net result of thus dusting the sample was to change the effective emissivity between sample and test plate, similar results might have been obtained by changing the emissivity of the hot plate, as in the present instance.

It is also possible that since the emissivity of the thermocouple used was much greater than the emissivity of the material tested, the thermocouples absorbed a disproportionate amount of radiant heat. However, with the thermocouples in contact with both hot plate and sample, the radiant energy available to them would, of necessity, have to be reflected by the surrounding fibers of the sample, and the effect thus produced is likely to be of little consequence.

Tests Nos. 11 and 12 were made in an attempt to determine the effect of radiation upon the individual sides of the sample. The copper insert plates were used on the hot side of the sample only, test No. 11 employing black lacquered surfaces while the highly polished plates were used for test No. 12. The results of these two runs were compared with tests Nos. 4 and 10, respectively, the difference being that these latter employed the insert plates on both sides of the sample.

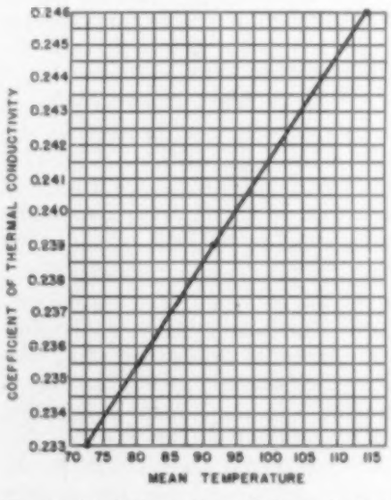

Fig. 4—Variation of coefficient of thermal conductivity with mean temperature

[4]Mechanism of Heat Flow in Fibrous Materials, by J. L. Finck. (*Bureau of Standards Journal of Research*, Vol. 5, Paper 243).

Very little difference in conductivity values of tests Nos. 4 and 11 was found, due to the fact that the highly oxidized condition of the cold plate produced an emissivity approaching that of the black lacquered plate. A comparison of tests Nos. 10 and 12 seems to indicate that by far the greatest surface influence (approximately 75 per cent) is felt on the hot side of the sample. However, the evidence is incomplete and no conclusion is drawn.

Summary

1. It must be remembered that the use of insert plates does not insure experimental accuracy and that, in particular, the use of a continuous plate for this purpose may introduce error. However, the purpose of this investigation was to secure comparative results only, which the authors believe has been accomplished.

2. For the samples used, a variation of the emissivity characteristics of the apparatus will cause a variation in the thermal conductivity values obtained. The greatest variation noted in these tests was approximately 38.5 per cent of the maximum conductivity value thus obtained.

3. It is suggested that approximately 75 per cent of the surface effects upon conductivity values occur on the hot side of the sample.

4. The observed increase in thermal conductivity values is not constant over a range of effective emissivities from 0.03 to 0.90, being greatest between effective emissivities of 0.80 to 0.90.

5. From the observed results it is suggested that either contact resistance or surface porosity, or both in combination, are responsible for the extreme variation encountered.

6. It is emphasized that the results obtained apply only to one material and one apparatus and are comparable only. No general conclusions are drawn or inferred except under these conditions.

From the results obtained by the use of this material it is suggested, and the authors believe it would be advisable, to perform additional experimentation to investigate the effect of plate surface characteristics upon a representative group of materials, including fibrous, porous, and smooth or paper covered or surfaced insulating materials. It is further suggested that a standard metallic hot plate be used for this work; that the plate emissivity be determined in the laboratory prior to each test, if necessary, in preference to the use of tabulated emissivity values from reference sources; that a smaller size of wire be used for the thermocouples; and that the thermocouples be imbedded in the surface of the sample, preferably concealed from the hot plate.

SOUTHERN CALIFORNIA MEMBERS MEET

March 12, 1941. The March meeting of the Southern California Chapter was held at Steven's Nikabob Cafe with 33 members and guests in attendance.

Through the courtesy of the Lockheed Aircraft the members and guests heard a very timely subject discussed, heating and ventilating of airplanes, which was rendered by A. E. Smith, and proved very educational. His talk was accepted with a great deal of interest, and he very clearly pointed out the problems faced by the manufacturers of airplanes.

This was followed by a picture on the growth and development of the Lockheed Plant, which was presented by K. W. Sneider.

February 13, 1941. The February meeting was held at the Walt Disney Studio, Burbank, Calif., with a total of 67 members and guests, who met at the Disney restaurant for dinner preceding the meeting.

After dinner the group was directed to one of the studios where the processing of animated pictures was done. A very clear, concise explanation of how the pictures are made and the difficulties which are encountered during the process was followed by an animated cartoon which was shown to the members and guests.

At this point in the program, Lou LeDeen, engineer for the Walt Disney Studios, gave a very interesting talk describing the air conditioning system, and the problems encountered in designing and installing the system to take care of the many unique problems in the processing of animated pictures.

RESEARCH PROJECTS DISCUSSED AT ATLANTA

February 19, 1941. The regular meeting of the Atlanta Chapter was called to order by Pres. S. W. Boyd at the Biltmore Hotel with an attendance of 65 members and guests. The minutes of the previous meeting were approved as read, and reports were submitted by the various committee chairmen, Messrs. Templin, Tucker, McKinney and Kent, the reports covering membership, research, finance, publicity. C. T. Baker reported that to date no bill had been introduced to revoke State Engineers' License Law, and further stated that reliable sources indicated that such a bill would not be introduced.

A short moving picture was then shown of the great naval battle between English and French ships, as well as of the collapse of the Tacoma Bridge, Tacoma, Wash.

T. T. Tucker then introduced the speaker of the evening, F. C. Houghten, director of ASHVE Research Laboratory, Pittsburgh, who gave a very interesting account of the work that has been carried out by the Laboratory, outlining the manner in which the Society determines what new projects are to be carried out by the Laboratory and how these tests are to be conducted. He described the functions of the Society's Research Committee, and called attention to the fact that H. King McCain, one of the Atlanta members, was recently elected to the Committee on Research for a three-year term.

From Mr. Houghten's talk, which he illustrated with slides, it was the opinion of the members present that the tests conducted at the Laboratory to date indicate that the problem of heat gain as it affects air conditioning design is more complex and more important than the problem of heat loss is in heating design. It was brought out that this is due to some extent to the fact that in the winter the outside temperature will only vary about 10 per cent in 24 hours, while in the summer the outside temperature will vary as much as 100 per cent. For this reason, Mr. Houghten stated that work must still be done on the insulating value and the heat capacity value of structures.

Following Mr. Houghten's talk he answered a number of questions from the floor, and President Boyd expressed appreciation to him for giving the Atlanta members the opportunity of hearing his interesting and instructive talk. According to A. H. Koch, secretary of the Chapter, the meeting adjourned after this discussion.

Heat Balance Tests of a Stoker-Fired Domestic Heating Plant

By R. S. Julsrud,* Harmon-on-Hudson, N. Y.

SUMMARY—Tests are reported for a domestic heating and air conditioning plant, fired by a bituminous bin feed ash removal stoker. To obtain good results with coals of different burning characteristics, the tests show that satisfactory fuel bed conditions can be obtained, provided attention is given to the careful study of the burning characteristics of each coal under consideration. From the standpoint of dependability the coking action of the coal in the retort is of major importance. No smoke nuisance was observed with any of the coals tested. High ash content is not objectionable, aside from the increased labor of removal. Observations indicated that a clean delivery and absence of dust in coal bin is desirable, and to fulfill this coal should be dust proofed.

THIS paper contains results of a series of heat balance tests on a domestic heating and air conditioning plant fired by a bituminous bin feed ash removal stoker. The object of these tests was to select a coal best suited to this heating plant based on the following criteria: (1) dependability, (2) availability, (3) cleanliness and, (4) economy. Dependability, meaning minimum attention, quick response to heat demands and uniform operation, is of major importance in a domestic heating plant. Once a dependable coal has been found a continued source of supply is important, particularly as in this case, in areas where a large selection of stoker coals is not available. Cleanliness of competing fuels has been a major obstacle to the domestic use of bituminous coal. Therefore, dust-proofing by oil or chemical means to insure a clean delivery and minimum dust during usage is desirable in domestic heating plants. Two factors determine the economy of a heating plant. They are: the delivered cost of fuel and the thermal efficiency of the heating plant. The annual fuel consumption of a domestic heating plant varying from 10 to 20 tons permits of fuel cost differences up to 20 per cent without seriously affecting the heating costs. However, thermal economy is usually affected by dependability, attention required, and uniform operation. Of the two economy factors, thermal economy is therefore more important than dollar economy in a domestic heating plant.

It is to be expected that the different burning characteristics of coals will affect their performance when fired and thereby their suitability for the job. For this heating plant it has been found that the following characteristics are of primary importance to successful results: (1) caking and/or coking tendencies; (2) density of coke formed on burning; (3) ease of ignition or reactivity; and (4) ash fusion temperature. Volatile content, ash content and coal sizing appear less important. During these tests the important burning characteristics were closely observed and are reported in the discussion of results.

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Description of House

The house (see Fig. 1) containing approximately 30,000 cu ft and eight rooms is constructed of 18-in. stone walls with clapboard gables. Six rooms are normally heated or about 25,000 cu ft of contents. The roof is slate covered and insulated with flax-compound insulation blocks 2 in. thick. The side walls have 2-in. air spaces between studs on 18-in. centers.

Fig. 1—View of house

Description of Heating Plant

The heating plant (see Figs. 2 and 3) consists of a cast iron furnace, steel incased warm air heater rated at 120,000 Btu per hour output, equipped with adjustable drip valve humidifier and integral air conditioner unit with four steel wool type air filters. The air distribution fan is of the centrifugal type with double inlet, top discharge. The fan capacity is 2,000 cfm at ¼ in. static pressure and is driven by a ⅓ hp single phase a-c motor. Warm air is delivered to room registers totaling 704 sq in. through ducts and risers from a trunk line. All ducts are covered with two sheets of asbestos insulation. Cold air returns from each room lead to a trunk line and thence to the air conditioner. The control consists of a mercoid air-stat to regulate the *bonnet* air temperature to the rooms, a mercoid warm air limit control to stop the stoker motor should the air fan fail and thus build up excessive *bonnet* temperatures, and a manually operated switch to circulate cool basement air through the house in the summer. The hot water heating system consists of a 1 in. diameter red copper coil, built and installed by the writer, encircling the furnace shell at the fire door level and properly connected to a 60 gal capacity horizontal storage tank. A

Fig. 2—View of heating plant

water encased gas heater provides hot water from June to October.

A bin-feed, ash removal bituminous coal stoker rated at 20 lb coal per hour serves the heater: It has an 18 in. diameter retort making the ratio of direct cast-iron heating surface to retort area equal 48.95 to 1.75 sq ft or 27.5 to 1. The stoker consists of four connected mechanisms which are, in the order of coal movement from the bin to the ash receiver: (1) the coal collector, (2) the coal travel tube, (3) the burner and (4) the ash remover, as shown in Fig. 3, an isometric view of the heating plant. A ⅙ hp motor drives a high pressure forced draft fan from one shaft extension and from the other, a steel worm engaging a bronze worm wheel which drives the ash conveyor screw through a roller chain reduction. The coal screw is driven through another roller chain mounted on the ash conveyor screw-extension. The bin-feed collector rotates through bevel gears from the coal feed screw. A cam mounted on the ash conveyor screw drives the rotating grates through an eccentric and shaft with an attached pawl at the end meshing with teeth on the under side of the grate. The grates revolve slowly on large ball bearings, about the stationary retort. Ash sifts through the segmented grates to the collector below from which it is scraped and dropped into the ash screw. Thence it is conveyed to covered ash cans at the side of the stoker. The entire ash removal system operates under suction eliminating any ash dust nuisance.

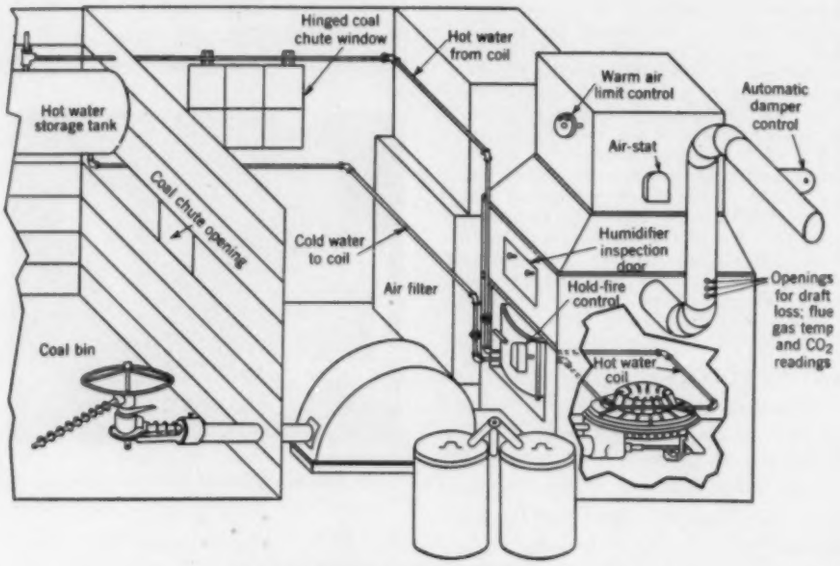

Fig. 3—Isometric view of heating plant

As coal passes up into the retort from the feed screw the volatile gases are rapidly evolved and ignite by radiation and conduction from the incandescent coke above the retort. The alloy steel retort is designed to admit air through the tuyeres at high velocities resulting in rapid ignition of the green coal to form coke masses immediately above the retort.

Test Procedure

Heat balance tests were run on three different coals on December 1, 7 and 15, 1940, beginning at 7:00 a.m. and ending at 7:00 p.m. The night before each test, the room thermostat was set back to 60 F (this being usual practice). With this setting the house temperature seldom falls below 60 F between 10:00 p.m. and 7:00 a.m. Thus the fuel bed is well burned down on natural draft at 7:00 a.m. when the thermostat is set up to 70 F. Prior to starting the tests, fly ash was removed from the circular *goose-neck* with a vacuum cleaner, the furnace hot water coil brushed clean, and the ash cans emptied. The rotating feed collector arm was removed and a barrel open at both ends was placed over the feed flight section of the feed screw. Initially 100 lb of coal were placed in the barrel and subsequent amounts added or withdrawn to empty the barrel at the conclusion of the test. An average coal sample was taken from small increments for later analysis. Ash discharged to the ash cans was weighed and a sample taken for carbon analysis. Fly ash on the heating surfaces was found to be very small and this loss was allowed for in the *unaccounted for* loss. Overfire and heater outlet draft was measured with an inclined tube draft gage and under-grate pressure with a U tube. Flue gas temperatures were obtained with an indicating thermo-couple previously calibrated with a mercury thermometer. Room and outdoor temperatures were taken with a combination temperature and humidity indicator. No CO readings were taken, as low volatile coals were being tested, and the furnace and stack gases gave no indication of smoke. Previous experience had shown marked variation in flue gas temperature and CO_2 per cent during *on* and *off* periods.

To obtain an over-all average for these quantities for heat balance purposes, it is necessary to obtain some measure of the relative amounts of coal burned in each period. To approximate these amounts the assumption made by Sherman and Cross[1] was used: namely, that the rate of burning is proportional to the square root of the pressure drop across the fuel bed. With this value and the average time of *on* and *off* operation, the weighted averages for per cent CO_2 and flue gas temperature were obtained. Power consumption for the stoker drive-motor and air distribution fan motor were averaged from several readings taken with the motors running separately and together with no other electric appliances in use. Operating data readings were taken at 15 min intervals and averaged for the 12 hour tests.

Discussion of Results

Table 1 gives operating data and heat balance results with three different coals tested. Data on expected *unaccounted for* losses are meager. Radiation from the heater

[1]Efficiency and Costs of Various Fuels in Domestic Heating, by Ralph A. Sherman and R. C. Cross. (*Technical Report No. 3*, Bituminous Coal Research, Inc., Washington, D. C.)

Table 1—Performance Data of Stoker Fired Domestic Heating Plant

Test No.		1			2			3		
Date of Test		12-1-40			12-7-40			12-15-40		
Duration of Test	hrs	12			12			12		
Fuel Data										
Seam		Upper Freeport			Bloss			Pocahontas No. 3		
State		Pennsylvania			Pennsylvania			West Virginia		
County		Somerset			Tioga			Mercer		
Location		Gray			Morris Run			Sagamore No. 1		
Size		— 2 in. x 0 in.			— ¾ in. x 0 in.			— 1¼ in. x 0 in.		
Dust Treatment		None			Oil Treated			Oil Treated		
Proximate Analysis (As Fired Basis)										
Moisture	%	3.5			2.3			2.8		
Volatile Matter	%	18.7			18.9			17.7		
Fixed Carbon	%	68.6			68.9			74.9		
Ash	%	9.2			9.9			4.6		
Heat Value per lb	Btu	13620			13480			14570		
Sulfur (Separately)	%	0.8			0.7			0.5		
Initial Deformation	F	2540			2520			2250		
Softening Temperature	F	2670			2780			2460		
Fluid Temperature	F	2800+			2900+			2625		
		"On" Period	"Off" Period	Total Period	"On" Period	"Off" Period	Total Period	"On" Period	"Off" Period	Total Period
Furnace Draft	IN. H_2O	0.05	0.10	.....	0.05	0.10	.....	0.02	0.04	.....
Draft Heater Outlet	IN. H_2O	0.06	0.11	.....	0.06	0.11	.....	0.03	0.05	.....
Forced Draft Pressure	IN. H_2O	1.2	0.00	.....	1.0	0.00	.....	0.80	0.00	.....
Heater Bonnet Temperature	F	.....	.....	200	.....	.....	200	.....	.....	200
Outdoor Temperature	F	.....	.....	36.5	.....	.....	32.0	.....	.....	34.2
Living Room Temperature	F	.....	.....	68.5	.....	.....	70.0	.....	.....	69.0
Boiler Room Temperature	F	.....	.....	60.0	.....	.....	62.0	.....	.....	58.0
Flue Gas Temperature	F	740	410	575	700	400	554	610	370	506
CO_2 Heater Outlet	%	10.2	7.8	9.2	11.0	7.2	9.2	10.8	8.5	9.70
Excess Air Heater Outlet	%	82.	145	102	69.	160	102	72.	120	92
Cold Water to Tank	F	.....	.....	52	.....	.....	49	.....	.....	51
Hot Water from Tank	F	.....	.....	120	.....	.....	119	.....	.....	117
Fuel Fired, Total	lb	.....	.....	150	.....	.....	140	.....	.....	115
Fuel Fired, per Hour	lb	.....	.....	12.5	.....	.....	11.6	.....	.....	9.6
Fuel Fired per Sq Ft G. S. per Hour	lb	.....	.....	7.0	.....	.....	6.5	.....	.....	5.4
Total Refuse	lb	.....	.....	15.5	.....	.....	15.0	.....	.....	6.9
Total Refuse (Per cent of coal burned)	%	.....	.....	10.35	.....	.....	10.7	.....	.....	6.0
Combustible in Refuse	%	.....	.....	35.0	.....	.....	32.0	.....	.....	28.0
Stoker Feed		.....	.....	.....	.....	.....	.....	.....	.....	.....
Average "On" Period	Min	35	.....	.....	40	.....	.....	32	.....	.....
Average "Off" Period	Min	.....	105	.....	.....	100	.....	.....	122	.....
Heat Balance										
Dry Gas Loss	%	.....	.....	16.50	.....	.....	15.90	.....	.....	13.80
H_2O in Air Loss	%	.....	.....	0.45	.....	.....	0.43	.....	.....	0.38
H_2O and H_2 in Fuel Loss	%	.....	.....	3.70	.....	.....	3.70	.....	.....	3.50
Carbon in Refuse Loss	%	.....	.....	3.88	.....	.....	3.71	.....	.....	1.72
Unaccounted for Loss	%	.....	.....	5.00	.....	.....	5.00	.....	.....	5.00
Total Losses	%	.....	.....	29.53	.....	.....	28.74	.....	.....	24.40
Overall Efficiency (By Difference)	%	.....	.....	70.47	.....	.....	71.26	.....	.....	75.6
Heating and Hot Water Costs										
Fuel Burned—Heating Season	Tons	.....	.....	11.5	.....	.....	11.5	.....	.....	10.0
Heat Value of Fuel—Heating Season	Therm	.....	.....	3130	.....	.....	3100	.....	.....	2914
Unit Fuel Cost per Ton	$	.....	.....	8.50	.....	.....	8.75	.....	.....	8.50
Unit Fuel Cost per Therm	$	.....	.....	0.031	.....	.....	0.033	.....	.....	0.029
Overall Efficiency	%	.....	.....	70.0	.....	.....	70.0	.....	.....	75.0
Heat Utilized—Heating Season	Therm	.....	.....	2190	.....	.....	2170	.....	.....	2175
Cost of Equipment										
Heating System	$	.....	.....	800	.....	.....	800	.....	.....	800
Stoker	$	.....	.....	200	.....	.....	200	.....	.....	200
Total Cost	$	.....	.....	1,000	.....	.....	1,000	.....	.....	1,000
Fuel Costs—Heating Season										
Fuel	$	.....	.....	98.00	.....	.....	100.60	.....	.....	85.00
Power for Air Distribution Fan	$	.....	.....	14.40	.....	.....	14.40	.....	.....	12.50
Power for Stoker	$	.....	.....	11.50	.....	.....	11.50	.....	.....	10.00
Total Fuel Costs	$	.....	.....	123.90	.....	.....	126.50	.....	.....	107.50
Maintenance—Heating Season										
Repairs to Heating System	$	.....	.....	5.00	.....	.....	5.00	.....	.....	5.00
Repairs to Stoker	$	.....	.....	15.00	.....	.....	15.00	.....	.....	15.00
Total Maintenance	$	.....	.....	20.00	.....	.....	20.00	.....	.....	20.00
Fixed Charges										
Interest on Equipment	$	.....	.....	30.00	.....	.....	30.00	.....	.....	30.00
Depreciation of Heating System	$	.....	.....	40.00	.....	.....	40.00	.....	.....	40.00
Depreciation of Stoker	$	.....	.....	20.00	.....	.....	20.00	.....	.....	20.00
Total Fixed Charges	$	.....	.....	90.00	.....	.....	90.00	.....	.....	90.00
Total Cost per Heating Season	$	.....	.....	233.90	.....	.....	236.50	.....	.....	217.50
Cost of Heat Utilized										
Based on Fuel Cost	$ per Therm	.....	.....	0.057	.....	.....	0.058	.....	.....	0.049
Based on Fuel and Maintenance	$ per Therm	.....	.....	0.066	.....	.....	0.067	.....	.....	0.058
Based on Total Cost	$ per Therm	.....	.....	0.106	.....	.....	0.108	.....	.....	0.100

casing is useful heat and should not be charged against the heating plant. *Unaccounted for* losses were placed at 5 per cent and based on: (1) fuel bed level error at start and finish of a test; (2) loss due to unburned soot and hydrocarbons deposited on the heating surface and lost to the stack; (3) errors in flue gas temperature and CO_2 readings. This arbitrary value of 5 per cent is believed to be generous for these losses.[2]

Test No. 1

This coal showed marked tendencies to *coke* or *mat* over the retort during the coking process. This blanket of plastic coal over the retort was slow to ignite and resistant to air flow through it. It was necessary to carry higher under-grate pressure and higher over-fire draft to hasten coking and burning up to reduce ash pit carbon losses. This

[2]Comparative Tests of Various Fuels When Burned in a Domestic Hot-Water Boiler, by E. S. Maloch and C. E. Baltzer. (Canada Department of Mines, *Bulletin* No. 705.)

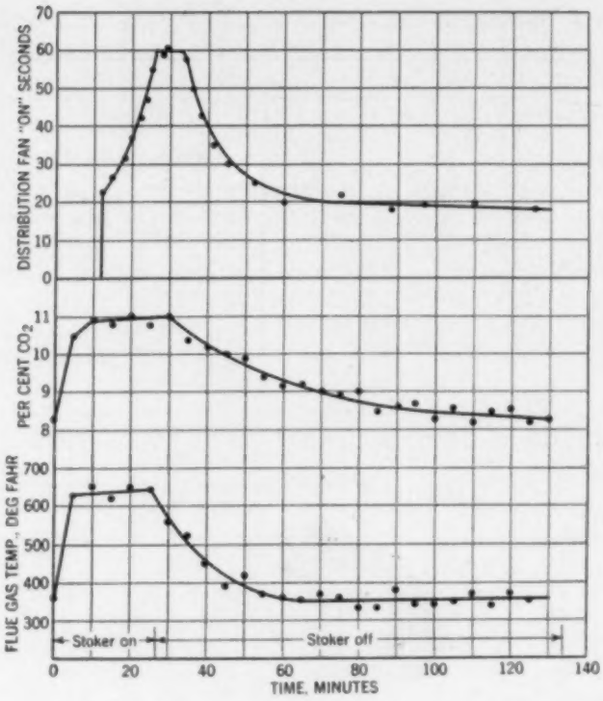

Fig. 4—Characteristic curves for a typical *on-off* cycle of stoker

increased the flue gas temperatures, reduced the CO_2 and created poor air distribution through the fuel bed. Clinker difficulties were experienced even though the coal analyzed 2800 F+ in fusion temperature. This is probably due to *blow-torch* action of the air issuing around the *matted* fuel in the retort creating very high temperatures as well as areas in a reducing atmosphere under the *matted* fuel. The top size of the coal was too large for handling by the feed flight of the feed screw resulting in irregular feeding. The dust nuisance of untreated coal was evident in the handling and movement of the coal.

The test showed that this coal would require considerable attention to maintain a well distributed fuel bed and thereby reduce clinker difficulties and also secure fair efficiency.

Test No. 2

This coal gave better performance and less attention than that of Test No. 1. However, the coal formed large, dense coke *spires* which, occasionally upon breaking at their base, fell clear of the retort leaving only a weak ring of flame surrounding the retort. After prolonged *off* periods (approximately 2 hours) these coke masses cooled, and unless pulled back over the retort would fail to ignite. This necessitated setting up the *hold-fire* control to maintain a higher furnace temperature and prevent loss of ignition. The high *ignition temperature* of this coke probably caused this difficulty. No clinker difficulties were experienced with this coal even though rather heavy fuel beds were carried to maintain high furnace temperature. Adequate oil treatment alleviated any dust nuisance. The high ash content was evident in the ash receiver cans.

The test showed that this coal would prove satisfactory where frequent heat demands prevailed resulting in maintaining a fairly high furnace temperature. During mild weather with long *off* periods, the house would probably be overheated resulting in fuel waste.

Test No. 3

This coal gave good thermal results and required a minimum of attention. While a strong coking coal, the coke *spires*, once formed, broke up into smaller pieces resulting in a well distributed fuel bed. This permitted carrying lower under-grate pressure and furnace draft with resultant lower flue gas temperature and higher CO_2. The coal responded quickly to heat demands. Once adjusted, no further changes in air or damper settings were required. While fairly clean, a more liberal treatment of oil would be desirable to entirely reduce dust.

No clinker difficulties were experienced, although the ash fusion (fluid) temperature of 2625 F, was some 200 F lower than the other coals. A porous, well distributed fuel bed, absence of localized hot zones and of coal *matting* account for this.

Fig. 4 shows curves of readings taken during test No. 3 which illustrate the rate of change of flue gas temperature; per cent CO_2 in flue gases, and the *on* periods of the air distribution fan during a typical *on-off* cycle of the stoker. During this time, the flue gas temperature and composition was taken at 5 min intervals and the *on-off* periods of the air circulating fan as they occurred. It will be seen that the flue gas temperature and per cent CO_2 rise rapidly to approximately their maximum value upon heat demand and remain quite constant while the stoker is on. Upon satisfaction of the heat demand flue gas temperature falls rapidly at first, then at a slower rate and finally levels off at a minimum value. The per cent CO_2 does not fall so rapidly nor so soon after the stoker stops and tapers off more slowly to its minimum value as excess air filters into the furnace from various sources and as the rate of burning slows up until a constant rate is acquired.

The top curve of Fig. 4 shows the response of the air distribution fan to heat demand. It will be seen that the rate of change of *bonnet* air temperature is slower than that of flue gas temperature or composition. This is due to the slower rate of heat transfer through the furnace shell. This time varies from 10 to 15 min normally, depending upon the fuel bed condition and temperature at the time of heat demand. Once the *bonnet* air temperature reaches the air stat setting of say 200 F the fan responds rapidly from approximately 20 seconds *on* and 180 seconds *off* to 60 seconds *on* and 60 seconds *off* at which heat transfer rate it becomes about constant. Upon the stoker stopping the *bonnet* cools more slowly than the flue gases, until a lower uniform heat transfer rate is established.

These curves show clearly the advantages of a solid fuel bed in establishing a more uniform room temperature than is possible with liquid or gaseous fuels. With such fuels, once the source of heat supply is shut off, the whole heating system cools rapidly resulting in a fluctuating temperature condition in the heated rooms. This applies likewise to the hot water supply in this case, though to a lesser extent, due to the difference in heat contents of the two fluids. This temperature lev-

elling out feature of solid fuel beds makes for more comfortable room conditions and should result in less fuel and power consumption than with liquid or gaseous fuels.

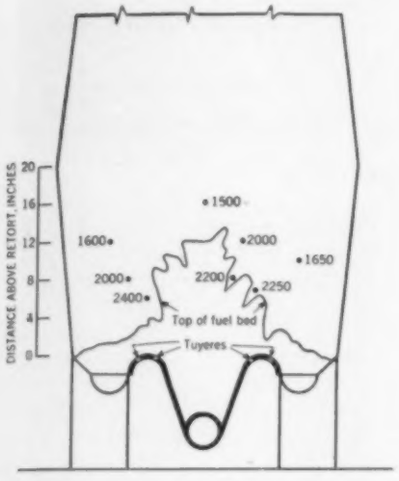

Fig. 5 — Furnace temperature readings taken through fire door with stoker *on*

(*Note*: Location of readings approximate only. Accuracy of readings approximate ± 50 F.)

Fig. 5 shows a few furnace temperatures taken with an optical pyrometer. The location of the readings with respect to the top of the fuel bed is approximate and, due to the continuous movement of the fuel, the temperatures may be in error ± 50 F. They were taken during a stoker *on* period to give some idea of the temperature gradients in this furnace.

Annual Heating and Hot Water Costs

A summary of these costs is shown in Table 1. The fuel burned per heating season was estimated as follows:

Tons Coal / Season = {Degree days (est.) × Total Register Area (sq in.) × Fuel Factor (lb 14000 Btu coal per sq in. Register Area per Degree Day)} ÷ overall efficiency

$$= \frac{5500 \times 704 \times 0.004}{\text{Efficiency}}$$

$$= \frac{16000 \text{ lb}}{\text{Efficiency}} \text{ approximately}$$

$$= \frac{8 \text{ Tons}}{\text{Efficiency}}$$

Comparison of consumption using this equation with coal burned during the 1939-40 heating season when both No. 1 and No. 2 test coals were used showed close agreement. No attempt has been made to segregate the heat utilized for heating the house, including humidifying water and that utilized to heat service water with the direct contact hot water coil here used. A rough check on daily hot water consumption showed about 75 gal used. On an eight month basis; 30 days per month: with 120 F — 50 F = 70 F rise and an assumed coil efficiency of 50 per cent gave 21,500,000 Btu required or about 10 per cent of the total heat utilized.

The heat value of fuel burned per season in therms is the product of the calorific value of coal per pound (as fired) × 2000 lb per ton × tons burned ÷ 100,000.

The unit cost of fuel per therm is the product of the cost per ton × 100,000 ÷ calorific value per ton (as fired).

The power costs are based on observed consumption of 25 kw per ton coal burned for the circulating air fan and 20 kw per ton coal burned for the stoker at an average cost of 5 cents per kilowatthour.

No charge was made for ash removal from the basement, this *chore* being done by the writer.

Maintenance costs are estimated. The heater and air conditioner have averaged $5.00 per season for circulating air fan motor trouble. Stoker repairs in two seasons have averaged $12.00 per season.

The interest charge on equipment is figured at 3 per cent. This is an approximate mean over a period when 6 per cent is charged on the undepreciated balance. Depreciation on the heating system is figured at 5 per cent on an expected 20 year life and on the stoker at 10 per cent on an expected 10 year life.

Conclusion

Subsequent use of Test No. 3 coal has shown that it approximates the criteria established in the introduction of this paper for coal selection for this heating plant; namely, (1) dependability, (2) availability, (3) cleanliness and (4) economy. The tests further show that it is possible to obtain good results with coals of different burning characteristics provided attention is given to studying these characteristics to obtain satisfactory fuel bed conditions.

The *caking* and *coking* action of the coal in the retort is of major importance from the standpoint of dependability, minimum attention and quick response and economy. Medium ash fusion coals with desirable coking properties should cause no clinker difficulties. No smoke nuisance was observed with any of the coals tested. Coals of higher volatile content would likely create no serious smoke troubles due to the rapid and intimate mixing of combustion air and combustible gases. High ash content, aside from the increased labor of removal, is not objectionable. Coal sizing, *viz.*, the percentage sizing through various mesh screens below 1½ in. does not affect performance. All coals tested were high in slack, that is, coal passing through a 20-mesh screen. A clean delivery and absence of dust in the coal bin is desirable and to fulfill this the coal should be dust proofed.

With these basic requirements for a coal for a domestic heating plant, a coal can be found that will fulfill the requirements.

PHILADELPHIA MEMBERS INSPECT COURT BUILDING

March 13, 1941. The regular monthly meeting of the Philadelphia Chapter of the Society was held at Hollands Restaurant and was called to order by Pres. C. B. Eastman.

F. D. Mensing introduced J. D. Cassell, Life Member of the Society, who gave an interesting short talk on the past and present status of the ASHVE, and called attention to the high standing of the Society in the scientific world of today.

The reading of the minutes of the February meeting was dispensed with on motion, duly seconded and carried.

About 8:00 p. m. the meeting was adjourned so the members could proceed to the Municipal Court Building, where arrangements were made to guide the Chapter in groups throughout the building, for inspecting the complete heating, ventilating and air conditioning system in this very modern Court Building. Considerable interest was shown in the many features of design and construction of the equipment.

Just a Reminder—California in June

Palace Hotel, San Francisco

June 16-20, 1941

ELABORATE plans are under way to greet Society members, their families and friends in San Francisco, June 16-20, when they arrive to attend the Semi-Annual Meeting 1941 with headquarters at the Palace Hotel. The Committee on Arrangements, under the guidance of G. M. Simonson, is working closely with local groups to make this Pacific Coast meeting one to remember. Assisting Mr. Simonson are Clyde Bentley and Dr. B. M. Woods as vice-chairmen. The chairmen of the various committees of Golden Gate Chapter are as follows:

Banquet.....N. H. Peterson
Entertainment.....C. L. Peterson
Finance.....A. J. Bouey
Ladies.....J. F. Kooistra
Publicity.....E. W. Simons
Reception.....J. A. Hill
Sports.....F. W. Kolb
Transportation.....R. A. Folsom

Numerous tours and inspections have been arranged, and there will be so many things to see and do in this beautiful country, time will be short, unless plans are carefully made in advance.

A transportation plan for the San Francisco Meeting of the Society has been developed, and special air conditioned cars will be operated *June 12* from Union Station, Chicago, over the Burlington Route. The train selected is the *Exposition Flyer,* operating on the following schedules:

FAST SCHEDULE
Via Moffat Tunnel and Feather River Canyon

Car B82—A. C. Standard Pullman Sleeper (12 Sec., 1 D.R.)
Car BX—A. C. Tourist Sleeper (16 Sec.)

June 12	12:35 p.m.	lv.	Chicago	lv.	CST	June 12
	2:45 p.m.	lv.	St. Louis	lv.	CST	
	11:25 p.m.	lv.	Omaha, Neb.	lv.	CST	
	8:00 p.m.	lv.	Kansas City	lv.	CST	
June 13	12:35 a.m.	lv.	Lincoln, Neb.	lv.	CST	June 13
	8:20 a.m.	ar.	Denver	ar.	MST	
	9:05 a.m.	lv.	Denver	lv.	MST	
	11:15 p.m.	ar.	Salt Lake City	ar.	PST	June 14
	11:25 p.m.	lv.	Salt Lake City	lv.	PST	
June 14	10:30 p.m.	ar.	San Francisco	ar.	PST	June 15

It should be noted that these two schedules provide an attractive trip in either standard Pullman or Tourist sleepers. The faster schedule via the Moffat Tunnel is for the convenience of those who must arrive in San Francisco for Committee Meetings, while the schedule via the Royal Gorge permits an early morning view of the Pike's Peak region, a brief stop in the Royal Gorge with its towering cliffs and roaring waters, then the Continental Divide, Eagle River Canyon, "Going to Sea by Rail" across the Great Salt Lake and a day along the route of the Forty-niners and through the High Sierras.

If a sufficient number (100 or more) selects one route which will permit running a special train from Salt Lake City, some additional sightseeing features will be added to the trip.

Members living east of Chicago can select the route they prefer and those from St. Louis, Kansas City, Minneapolis, Omaha and Des Moines, etc., can join their friends en route. For schedules connecting with the special train from Chicago confer with Local Ticket Agents and make arrangements for any return route desired. The following optional return routes are available via:

1. Ogden, Cheyenne, Omaha, Chicago.
2. Los Angeles, Grand Canyon, Carlsbad Caverns, Indian Detour, Albuquerque, Wichita, Kansas City and St. Louis, or Chicago.
3. Los Angeles, Phoenix, El Paso: (*a*) San Antonio, New Orleans, Atlanta, Washington; (*b*) Kansas City and Chicago; (*c*) Fort Worth, Dallas and St. Louis.
4. North Pacific Coast—Portland, Seattle, Spokane, Yellowstone Park, St. Paul, Chicago.
5. North Pacific Coast—Portland, Seattle, Spokane, Glacier Park, St. Paul, Chicago.
6. Portland, Seattle, Victoria, Vancouver, Lake Louise, Banff, St. Paul, Chicago.
7. Portland, Seattle, Vancouver, Jasper Park, Winnipeg, St. Paul, Chicago.
8. Los Angeles, Boulder Dam, Utah Parks (Bryce-Zion), Salt Lake City, Royal Gorge, Colorado Springs, Denver: (*a*) Kansas City and St. Louis; (*b*) Omaha and Chicago.

SCENIC ROUTE
Via Royal Gorge and High Sierras

Car B17—A. C. Standard Pullman Sleeper (12 Sec., 1 D.R.)
Car BH—A. C. Tourist Sleeper (16 Sec.)

	12:35 p.m.	lv.	Chicago	lv.	CST	June 13
	2:45 p.m.	lv.	St. Louis	lv.	CST	
	11:25 p.m.	lv.	Omaha, Neb.	lv.	CST	
	8:00 p.m.	lv.	Kansas City	lv.	CST	
	1:20 a.m.	lv.	Lincoln, Neb.	lv.	CST	June 13
	8:20 a.m.	ar.	Denver	ar.	MST	
	8:40 a.m.	lv.	Denver	lv.	MST	
	7:30 a.m.	ar.	Salt Lake City	ar.	PST	June 14
	7:50 a.m.	lv.	Salt Lake City	lv.	PST	
	8:25 a.m.	ar.	San Francisco	ar.	PST	June 15

PALACE HOTEL RATES

Single rooms with bath—
$3.50, $4.00, $5.00, $6.00 and $7.00 per day per person
Double rooms with bath—
$6.00, $7.00, $8.00 and $9.00 per day (2 persons)
Twin-bed rooms with bath—
$7.00, $8.00, $9.00 and $10.00 per day (2 persons)
Suites—
$10.00, $15.00, $20.00 per day.

With these routes in mind and the excellent technical sessions which have been arranged, one cannot afford to let such an opportunity pass by. Another advantage the Society members will have, will be the opportunity to attend the 52nd Annual Convention of the *Heating, Piping and Air Conditioning Contractors National Association,* which will have its headquarters at the St. Francis Hotel, and will convene the week of June 16. The Association has planned a meeting of great importance to their industry, presenting problems which must be faced and solved, especially problems arising under the National Defense Program.

A joint session will be held with the *HPACCNA* (see Schedule of Events p. 336) at the St. Francis Hotel, on Monday morning, June 16, and will have W. L. Fleisher, President of the ASHVE, as the presiding officer. This will be followed by a joint luncheon at the St. Francis where the members of both organizations can enjoy a friendly get-together.

Topics of interest to all will be discussed at the Society's technical

sessions opening Tuesday and among these subjects are panel heating, weather data, conservation of underground water, application of gas fuel to heating boilers, thermal conductivity of wood, effect of insulation on plant performance, heat losses, thermodynamic properties of air and water, effect of local cooling on reactions of individuals, and a resumé of research in cooperative institutions. In connection with research, it is planned to have an inspection trip to the University of California where research work is now under way.

Those planning to be in California the week of June 16 will not only have the Society's Semi-Annual Meeting and the Annual Convention of the *HPACCNA* to attend, but they will have an opportunity to spend many hours at the Civic Auditorium, where the Pacific Heating and Air Conditioning Exposition will be held from June 16 to 20. Here one will be able to view the very latest in modern apparatus for heating, ventilating and air conditioning installations. C. F. Roth, manager of the Exposition, has promised an outstanding event, and he has announced that more than three-fourths of the exhibition space has already been reserved.

Mark the calendar and make plans for a busy time in San Francisco during the week of June 16.

Clyde Bentley

G. M. Simonson

B. M. Woods

N. H. Peterson

R. A. Folsom

J. A. Hill

F. W. Kolb

Round Trip Railroad Fares to San Francisco

From	Coach Class	First Class*	1st Class Intermediate	Coach* Intermediate	Round Trip Lower Berth Standard Pullman
Atlanta, Ga.	$74.60	$106.70	$ 91.20	$ 83.10	$36.80
Baltimore, Md.	89.70	132.20	116.70	98.20	43.05
Boston	97.20	145.65	129.35	105.85	46.75
Buffalo	86.10	120.65	104.35	95.10	40.45
Chicago	65.00	90.30	74.00	.....	33.10
Cincinnati	74.30	105.45	89.95	82.80	36.75
Cleveland	78.70	110.00	95.70	87.80	37.80
Dallas	54.40	75.60	61.95	.....	26.80
Denver	43.50	60.40	49.45	.....	23.10
Des Moines	58.75	81.55	66.85	.....	29.95
Detroit	76.40	106.75	90.45	85.40	37.80
Durham, N. C.	85.50	124.65	109.15	94.00	45.20
Grand Rapids	72.40	101.00	84.70	81.40	37.30
Greensboro, N. C.	83.70	121.80	106.30	92.20	43.50
Houston	54.40	75.60	61.95	.....	26.80
Indianapolis	71.70	99.95	84.45	80.20	35.70
Kansas City	54.40	75.60	61.95	.....	27.85
Los Angeles	10.80	19.90	16.00	.....	6.30
Louisville, Ky.	70.50	99.85	84.35	79.00	35.70
Madison, Wis.	65.00	90.30	74.00	.....	33.10
Memphis	64.40	89.25	73.30	.....	30.45
Milwaukee	65.00	90.30	74.00	.....	34.70
Minneapolis	65.00	90.30	74.00	.....	33.10
Montreal	98.20	140.55	124.25	.....	45.70
Nashville	70.50	99.85	84.35	79.00	33.10
Newark, N. J.	90.00	135.00	123.10	101.60	45.00
New Haven	90.00	135.00	123.45	101.85	45.00
New Orleans	64.40	89.25	73.30	.....	29.40
New York	90.00	135.00	123.45	101.85	45.00
Norfolk	91.25	132.85	117.35	99.75	43.05
Omaha	54.40	75.60	61.95	.....	25.10
Oklahoma City	54.40	75.60	61.95	.....	26.80
Philadelphia	90.65	135.50	119.20	99.15	44.10
Pittsburgh	83.80	117.40	101.10	92.80	39.40
Portland	22.50	33.70	25.30	.....	11.60
Racine, Wis.	65.00	90.30	74.00	.....	33.10
Richmond, Va.	89.40	129.45	113.95	97.90	43.05
Seattle	26.50	41.95	33.15	.....	14.20
St. Louis	61.60	85.60	85.60	.....	31.50
Toledo	74.40	103.85	87.55	83.40	37.80
Toronto	87.25	119.85	103.55	.....	40.95
Utica, N. Y.	90.00	130.85	114.55	98.50	43.10
Washington	89.70	132.20	116.70	98.20	43.05
Wichita, Kans.	54.40	75.60	61.95	.....	25.20
Winnipeg	80.25	108.20	91.40	.....	40.45
Worcester, Mass.	95.40	143.05	126.75	104.05	46.80

*Pullman and Tourist Sleeper charges are additional.

A. J. Bouey

J. F. Kooistra

C. L. Peterson

E. W. Simons

PROGRAM

Semi-Annual Meeting, 1941

American Society of Heating and Ventilating Engineers

Palace Hotel—San Francisco, Calif.

June 16-20, 1941

Sunday, June 15

10:00 a. m. ASHVE Registration—Palace Hotel
2:00 p. m. Committee Meetings

Monday, June 16

8:30 a. m. ASHVE Registration—Palace Hotel.
10:00 a. m. Joint Session with *Heating, Piping and Air Conditioning Contractors National Association* — St. Francis Hotel—W. L. Fleisher presiding.

(*a*) Invocation
(*b*) Singing "America"
(*c*) Greetings by City Official
(*d*) Response by J. E. McNevin, President, *HPACCNA*
(*e*) Greetings by Local Presidents ASHVE and *HPACCNA*
(*f*) Program Announcements
(*g*) Group photograph
(*h*) Speaker

12:15 p. m. Joint Luncheon—St. Francis Hotel (sponsored by *HPACCNA*)
2:00 p. m. Opening of Pacific Heating and Air Conditioning Exposition—Civic Auditorium
2:30 p. m. Council Meeting
7:00 p. m. Dinner, Entertainment and Dance—St. Francis Hotel (sponsored by *HPACCNA*)

Tuesday, June 17

1:30 p. m. TECHNICAL SESSION
Panel Heating and Cooling Analysis, by B. F. Raber and F. W. Hutchinson
Weather Data for Cooling Design Application, by J. D. Kroeker
Conservation of Underground Water with Suggestions for Control, by N. E. Porter
12:00 p. m. Exposition—Civic Auditorium
7:00 p. m. Sightseeing

Wednesday, June 18

10:00 a. m. TECHNICAL SESSION
The Application of Gas Fuel to Existing Heating Boilers, by R. L. Grutzmacher
The Thermal Conductivity of Wood, by J. D. MacLean
Effect of Insulation on Plant Performance in the Research Residence, by A. P. Kratz and S. Konzo
An Analysis of Factors Influencing Building Heat Losses, by P. D. Close
12:00 p. m. Exposition—Civic Auditorium
7:00 p. m. Banquet and Dance (sponsored by ASHVE)

Thursday, June 19

10:00 a. m. TECHNICAL SESSION
Resumé of Society Research in Cooperative Institutions, by A. E. Stacey, Jr.
The Interaction Constant for Moist Air, by J. A. Goff and A. C. Bates
Effect of Local Cooling on Reactions of Individuals in Various Environments, by F. C. Houghten
12:00 p. m. Exposition—Civic Auditorium
2:00 p. m. Inspection Trip to University of California

Friday, June 20

ASHVE—JOINT PROGRAM—*HPACCNA*
Sightseeing and Inspection Trips
Deep Sea Fishing Contest (ASHVE *vs.* *HPACCNA*)
Golf Tournament (ASHVE *vs.* *HPACCNA*)
Exposition—Civic Auditorium

SCHEDULE OF EVENTS

52nd Annual Convention

HEATING, PIPING AND AIR CONDITIONING CONTRACTORS NATIONAL ASSOCIATION

Headquarters—Hotel St. Francis,
San Francisco, Calif.
June 16-20, 1941

Sunday, June 15

10:00 a. m. Meeting of Board of Directors

Monday, June 16

8:30 a. m. Registration
10:00 a. m. Joint Session
12:15 p. m. Joint Luncheon with ASHVE.
2:00 p. m. Opening of Pacific Heating & Air Conditioning Exposition
7:00 p. m. Annual Dinner and Dance

Tuesday, June 17

9:30 a. m. Convention Session
12:30 p. m. Luncheon for Advertisers in Official Bulletin
7:00 p. m. Roundtable Discussions
Exposition and Sightseeing

Wednesday, June 18

2:00 p. m. Convention Session
7:00 p. m. Members invited to attend Banquet of the ASHVE

Thursday, June 19

9:00 a. m. Convention Session
2:00 p. m. Inspection Trip to University of California
Meeting of Board of Directors
7:00 p. m. Exposition

Friday, June 20

Joint Entertainment and Sightseeing Program

WISCONSIN MEMBERS ENTERTAIN ASRE PRESIDENT

March 14, 1941. The Wisconsin Chapter met for its monthly meeting at the Knickerbocker Hotel, which was held jointly with the *American Society of Refrigerating Engineers*, and 66 members and guests attended.

Pres. A. S. Krenz called attention to the Students' Exhibit at the Mechanical Engineers' Building at Madison, and invited all those present to attend.

Ernest Szekely suggested that the Society consider the revision of the Heating and Ventilating Code, and this opinion was concurred in by H. C. Frentzel. C. H. Randolph, J. H. Volk and Carl Becker led the discussion regarding the proposed revisions to be made to the code to bring it up to date. E. W. Gifford then made a motion to have the Society appoint a committee to study the revision of the present Heating and Ventilating Code and report back to the membership, and the motion was duly seconded and carried.

President Krenz then turned the meeting over to Mr. Tokach, chairman of the program committee for the *ASRE*, who introduced the first speaker of the evening, W. S. Moehlenpah, air conditioning engineer, Chicago, Milwaukee, St. Paul and Pacific Railroad. The speaker addressed the gathering on railroad air conditioning, its installation, operation and maintenance, which he illustrated with slides. His talk was gratefully received. He omitted the elementary principles and took up the subject particularly from the angle in which it differed from the ordinary comfort cooling and air conditioning jobs.

The principal speaker of the evening was L. L. Lewis, vice-president and chief engineer, Carrier Corp., Syracuse, N. Y., also National President of the *ASRE*, who spoke on *What An Engineering Society Offers You*. Mr. Lewis stressed the benefits which accrue to engineers belonging to an engineering society and recounted from his personal experiences some of the benefits that he has received over a period of many years.

Upon completion of Mr. Lewis' talk, Mr. Tokach then called for a rising vote of thanks to both speakers, which was followed by adjournment, as reported by H. W. Schreiber, secretary of the Chapter.

LATENT AND SENSIBLE HEAT DISCUSSED AT ATLANTA

March 7, 1941. The March meeting was held at the Biltmore Hotel, and was called to order by President Boyd, with 31 members and guests present. The minutes of the previous meeting were approved as read.

Following Mr. Templin's report on membership, T. T. Tucker, chairman of research committee, reported that a meeting was held while Director Houghten of the Research Laboratory was still in town, and, while Dr. Houghten injected several new thoughts into the outline of the research program, it was finally decided to let the program remain as initially set up. Mr. Tucker further advised that the final form of the contract would be signed shortly.

President Boyd introduced the guest of the evening, Scott F. Nicoll, mechanical engineer, York, Pa., who selected as his subject Reheating by Means of Refrigerant Compressor Discharge Gas. He prefaced his talk by defining the various words and phrases used in presenting his subject. Following this he gave an excellent description of how latent and sensible heat removal can be accomplished at the same time with the same apparatus. He supplemented his talk by picture slides which outlined the several ways in which reheat could be introduced into a cooling system and from data gathered from several actual jobs, he showed through a process of constructive steps, the most satisfactory system which could be set up to meet the sensible and latent heat requirements. These figures were then applied to three different types of bypass systems in which the refrigerant hot gas was employed as a means of reheat so as to balance out both the latent and sensible heat requirements for these three types of systems.

This was a very interesting presentation as was evidenced by the number of questions which were asked by the audience.

Following Mr. Nicoll's talk, Mr. Tucker, in the absence of Mr. Kent, gave a resumé of the National Chapter Delegates Meeting.

Mr. Tidmore, smoke inspector, outlined the progress made on the heating code, his talk being supplemented a great deal by C. B. Cole. A number of questions were asked regarding certain phases of the code, following which it was decided to continue the discussion at the next meeting of the Chapter. Adjournment followed at 10:00 p.m.

MANITOBA MEMBERS HAVE A QUESTION BEE

January 23, 1941. The regular monthly meeting of the Manitoba Chapter of the Society was held in the Fort Garry Hotel, Winnipeg, with Pres. P. L. Charles presiding. Dinner was served preceding the meeting which was called to order by the president. The reading of the minutes of the previous meeting was the first order of business, and these were adopted on motion of E. M. Hepburn, seconded by J. R. Stephenson.

G. C. Davis reported for the entertainment committee that the annual party, originally planned for January, had been postponed. There being no other committee reports, President Charles advised that a request had been received for questions to be discussed, and a lively discussion resulted in the submission of the following questions:

1. What plans has the Speakers Bureau for furnishing speakers to remote Chapters such as Manitoba?
2. Could the Guide Publication Committee publish a supplement to THE GUIDE, in loose-leaf form of a convenient size, containing tables from THE GUIDE which are in constant use?
3. Has the Society any data on the increase in pressure in a closed vessel containing a liquid when a temperature rise occurs? In other words, what is the rise in pressure (total and per degree) in a range boiler from 50 F and 40 lb originally, to 200 F?
4. Under what conditions does the Society make a grant to a University for research?

Those submitting the questions were Messrs. Davis, Michie and Steele.

Considerable discussion followed, and after arrangements for the an-

nual party were completed, the secretary was instructed to bill all chapter members for 1941 chapter dues and to show 1940 arrears where such were outstanding.

Motion was made by J. B. Steele, seconded by Mr. Davis, and carried, that if any record existed of arrears of chapter dues for 1939 or earlier, such record be closed. Following this the meeting adjourned at 9:10 p.m.

ONTARIO ANNOUNCES CANDIDATES

March 3, 1941. There were 70 members and guests in attendance at the March meeting of the Ontario Chapter which was held at the Royal York Hotel. Vice-Pres. C. Tasker officiated in the absence of Pres. J. W. O'Neill. Following dinner Mr. Tasker introduced the visitors from Winnipeg and Hamilton, and announced that the next regular meeting would be held on April 21, at which time the Council of the Society would be present.

M. W. Shears, chairman of the nominating committee announced the names of candidates for the coming year as follows: *President*—C. Tasker; *Vice-President*—D. O. Price; *Secretary-Treasurer*—H. R. Roth; *Board of Governors*—W. C. Kelly, E. R. Gauley, A. S. Morgan, and J. W. O'Neill who automatically serves as retiring president.

D. O. Price then addressed the meeting on gravity warm air, followed by a talk on forced warm air which was given by H. G. Hill. R. W. Bayles followed with a talk on gravity hot water, and A. T. Jones concluded the discussion with a talk on forced hot water.

R. H. Lock extended a vote of thanks to the speakers following a round table discussion, which was followed by adjournment.

NATURAL GAS AND AERONAUTICS TOPICS AT MINNESOTA

March 3, 1941. The regular monthly meeting of the Minnesota Chapter was held at the Coffman Memorial Union, on the Campus of the University of Minnesota, with Pres. M. J. Bjerken presiding. The minutes of the February meeting were approved as read by the secretary. President Bjerken, who represented the Minnesota Chapter at the Society's Annual Meeting in Kansas City, submitted his report on the meeting.

The speaker of the evening was Lester Eck, General Superintendent of the Minneapolis Gas Light Co., who discussed the production and utilization of natural and manufactured gas. He explained how natural gas is brought to Minneapolis by pipeline from Texas. Before distribution, the natural gas is mixed with manufactured gas and conditioned by mixing with steam and oil. Mr. Eck further discussed the significance of variations in peak loads over a period of years. It was the consensus of opinion that this was one of the most interesting meetings of the year, and Mr. Eck's talk was most satisfactorily received. The meeting adjourned at 9:00 p. m.

February 3, 1941. The February meeting of the Minnesota Chapter was held at the Coffman Memorial Union on the University of Minnesota Campus, with 35 members and guests in attendance.

Prof. F. B. Rowley, R. E. Backstrom and H. E. Gerrish discussed the work being done in the promotion of a new combined Mechanical and Aeronautical Building at the University. At the suggestion of Mr. Gerrish, the Chapter voted that the secretary direct a letter to the State Legislature urging the support of members of that body in obtaining funds for this building.

The speaker for the evening was Prof. John Akerman, head of the Aeronautical Department at the University of Minnesota. Professor Akerman gave a very interesting discussion of the problems in connection with high altitude flying. He told of his experiences in developing an improved method of dispensing oxygen at high altitudes. According to Professor Akerman, one of the biggest problems in aviation today is the heating and ventilating of planes. After considerable interesting discussion the meeting adjourned at 9:15 p. m., according to D. B. Anderson, secretary of the Chapter.

W. L. FLEISHER MOVES TO NEW QUARTERS

Walter L. Fleisher, President of the Society, announced the removal of his office from 11 West 42nd St. to 475 Fifth Ave., New York, N. Y., on April 10.

HACH JOINS LABORATORY STAFF

E. C. Hach, Westfield, N. J., has been appointed to the staff of the ASHVE Research Laboratory located at the Experiment Station of the U. S. Bureau of Mines, Pittsburgh, Pa., as announced recently

E. C. Hach

by A. E. Stacey, Jr., Chairman of the Committee on Research. Mr. Hach has an extensive background of experience in which he has served as Chief Engineer of the Manufacturing Department of Standard Air Conditioning, Inc., a division of American Radiator & Standard Sanitary Corp., New York, where he was in charge of research development and product design.

Prior to this connection he was employed by the Carrier Corp., Syracuse, N. Y., where he was engaged in research and development work both in the laboratory and in the field, and he also has held positions with the Beckwith Co., Dowagiac, Mich., Western Electric Co., Cicero, Ill., and the Mutual Fire Prevention Bureau, Chicago, Ill.

Mr. Hach, born in Philadelphia, was graduated from Michigan State College in 1920 with a B.S. in Mechanical Engineering; he is a registered professional engineer in the State of New York; a member of Tau Beta Pi, Honorary Engineering Fraternity, and since 1939 has been a member of the American Society of Heating and Ventil-

ATING ENGINEERS. He is the author of a number of publications including Regulations for the Installation of Oil Burning Equipment, and has prepared numerous engineering manuals and pamphlets on the subject of heating, ventilating and air conditioning.

ALFRED KELLOGG, LIFE MEMBER, DIES AT 75

Alfred St. Claire Kellogg, consulting engineer of Boston, Mass., died at his home, 12 Wiley Road, Belmont, Mass., on April 16, 1941. Mr. Kellogg was born in Danbury, Conn., in 1866, receiving his early education in the local schools and then attended Harvard University.

From 1889 to 1893 he was connected with Fuller and Warren Co., as superintendent and engineer, and in 1899 was employed by the American Blower Co. In 1900 he became part owner in the Massachusetts Fan Co. where he remained until 1907 when he established his own business as consulting engineer in power, heating, lighting, refrigeration and air conditioning.

Mr. Kellogg joined the ASHVE in 1916 and always took an active part in its affairs. Since 1936 he has been a Life Member of the Society, a privilege and honor enjoyed by only a limited group.

He was the first president of the Massachusetts Chapter and served on its Board of Governors continuously from 1919 to 1926, and was on the board at the time of his death. He served on the Society's Council in 1920, 1921, 1923 and 1924, on the Membership Committee in 1920 and 1921, and as chairman of that committee in 1924. In 1922, 1925 and 1926 he served on the Nominating Committee, and as a member of the Committee on Research for a period of eight years, from 1919 to 1926.

He was a member of the Committee on Furnace Heating Installation in 1918, and a member of the Committee on Chapter Relationship, and the Committee on Legislation in 1919. In 1921 he served as a member of the Committee on Revision of Constitution and By-Laws, and as Chairman of the Committee on Chapters in 1921, 1923 to 1927.

In 1922 he acted as one of the consulting engineers on the Committee on Steam and Return Main Sizes, and as a member of the Publication Committee in 1923. In 1925 he served as a member of the Committee on Code of Ethics, and that same year served as chairman of the Committee to Draft a Code for Guidance of Membership Committees. In 1926 and 1927 he was a member of the Committee to Cooperate with the Rochester School Board. He was appointed chairman of the Committee on Rating Low-Pressure Boilers in 1927, and in 1928 served on the Committee on Maximum Boiler Output, and as chairman of the Committee on Rating and Selection of Low-Pressure Heating Boilers.

Alfred Kellogg

Mr. Kellogg, as well as being a member of the ASHVE, was also a member of the *American Society of Mechanical Engineers* and the *Illuminating Engineering Society*. His passing will be a great loss to his many friends and associates in the engineering field.

The Officers and Council of the Society extend their sincerest sympathy to his widow, who survives.

PRESLY M. O'CONNELL DIES IN SEATTLE

Presly M. O'Connell died at his home, 5749 31st Ave., Seattle, Wash., on March 9, 1941, at the age of 53. He was born in Pittsburgh, Pa., July 3, 1888, and received his preparatory education in the public schools there. He served as an apprentice with Baker-Smith Co., from 1902 to 1906, at which time he was employed by the American Warming and Ventilating Co., Pittsburgh, where he received his drafting experience. He remained with them until 1909, when he went with the Columbus Heating and Ventilating Co., Columbus, O., working there in the shop until 1912. From 1912 to 1914 he was on the staff of Lewis and Kitchen, Chicago, Ill., and later became associated with Lewis and Capron Co., also in Chicago, as engineer.

Prior to his employment with Lewis and Capron he had 13 years of testing experience in designing, installation and operating large heating and ventilating systems. He had made a life work of the designing of heating and ventilating apparatus for public school buildings.

In 1918 Mr. O'Connell was employed as heating engineer by William K. Robertson, Consulting Engineer, Minneapolis, Minn., at which time he was engaged to complete the engineering work at the Base Hospital, Camp Dodge, Ia. This work brought about his writing a paper entitled "Heating a Base Hospital," which was published in the JOURNAL of the Society, in which he described the system installed in the cantonment, consisting of 70 buildings, with an aggregate amount of 145,000 sq ft of radiation, all taken care of by one central heating plant of fourteen 150 hp boilers.

The latter part of 1918 he was called into service in the U. S. Army, and was assigned to the Construction Division at Camp Dodge, where he had charge of the heating systems installed. At the completion of his work at Camp Dodge he established himself at Sioux Falls, S. D., where he was connected with Perkins and McWayne. Here he was in charge of the installation in such buildings as the High School, YMCA Building, a local department store costing $200,000, and several other school buildings costing approximately $90,000.

In 1924 he became associated with John Graham, architect, Seattle, Wash., as sanitary and mechanical engineer, and later became mechanical engineer for the Seattle School District. In 1934 he received an appointment as resident engineer inspector for the PWA, located in Pullman, Wash. He returned to Seattle in 1937, retaining his position as engineer inspector for the Public Works Administration.

Mr. O'Connell was very active in the work of the Pacific Northwest

Chapter of the Society, serving as its president in 1933, and on its various committees from time to time.

The Officers and Council of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS extend sincerest sympathy to his widow and family who survive.

E. F. GLORE DIES

Evins F. Glore, president of the Evins F. Glore Sales Corp., New York, N. Y., dealers in heating equipment, died at his home, 644 Riverside Drive, April 12, 1941.

Mr. Glore was born in Louisville, Ky., in 1873, and came to New York in 1889, at which time he became associated with the American Radiator Co. as salesman and branch manager, which position he held until 1907. In 1911 he went with Richardson and Boynton Co., New York and Philadelphia, as manager and sales manager, remaining with them until 1913. In 1914 he was appointed manager of Pierce, Butler and Pierce Mfg. Co., New York. Later he was also associated with the Syracuse Faucet and Valve Co., the Abram Cox Stove Co., Philadelphia, and in 1931 became president of the Republic Radiator and Mfg. Corp., Baltimore, Md. In 1932 he established the Evins F. Glore Sales Corp., with offices in the Grand Central Terminal Bldg., New York, where he remained as its president until his death.

Mr. Glore was a veteran of the old 7th Regiment of the New York National Guard, now the 207th Coast Artillery (anti-aircraft), and was one of the oldest members of the West Park Presbyterian Church, New York, and was a former member of the choir of the Broadway Tabernacle.

He joined the ASHVE in 1916, remaining on its rolls for a period of over 22 years. The Officers and Council of the Society extend their sincere sympathy to his widow, Mrs. Alberta R. Glore, who survives.

HERBERT A. TERRELL DIES

Herbert A. Terrell, retired heating engineer, died March 26, 1941, at his home, 301 Springfield Ave., Cranford, N. J., where he had made his home for a number of years.

A native of Indiana, Mr. Terrell was graduated from the Massachusetts Institute of Technology in 1906, and served with Warren Webster and Co., Camden, N. J.; the Atmospheric Conditioning Co., Philadelphia, and the Carrier Engineering Corp., when in Newark, N. J.

Mr. Terrell was a member of the ASHVE for a period of 18 years when he retired from active business in 1933.

He is survived by his wife, Mrs. Dorothy Terrell; a daughter, Miss Lois Terrell, and two sisters, Miss Nellie E. Terrell, Trenton, N. J., and Mrs. Jay Williams, Los Angeles, Calif., to whom the Officers and Council of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS extend their sincere sympathy.

Candidates for Membership

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the JOURNAL of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Committee on Admission and Advancement as soon as possible.

When the Committee on Admission and Advancement has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 48 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Committee on Admission and Advancement, and in turn, the Council, urge the members to assume their share of responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some member by May 15, 1941, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
ALLISON, ROBERT E., Owner, American Sheet Metal Co., Dallas, Tex.	C. R. Gardner M. L. Brown	S. H. Green L. S. Gilbert
BABCOCK, PAUL R., Designer, G. M. Simonson, Cons. Engr., San Francisco, Calif.	J. Gayner G. M. Simonson	N. H. Peterson C. E. Bentley
BOGARDUS, GEORGE W., Br. Mgr., Kewanee Boiler Corp., Des Moines, Ia. (*Reinstatement*).	B. E. Landes R. H. Schnell	E. H. Borg T. R. Johnson
CARROLL, DANIEL E., Pres., Carroll Sheet Metal Works, Inc., Woodside, L. I., N. Y.	D. D. Kimball G. A. Belsky	I. Willner J. A. Heller
CHAPMAN, D. B., Dist. Sales Mgr., Clarage Fan Co., Kalamazoo, Mich.	J. M. Rittelmeyer (*Non-member*) T. T. Tucker	S. W. Boyd C. L. Templin
CHASE, ROGER E., JR., Mfrs. Agent, Portland, Ore.	J. D. Kroeker T. E. Taylor	B. W. Moore C. W. Brissenden
CREW, FRANCIS D., Pres., The F. D. Crew Co., Philadelphia, Pa.	E. C. Powers E. K. Wagner	L. A. Childs E. Elliot
DEMING, ROY E., Htg. Engr., Premier Furnace Co., Dowagiac, Mich. (*Advancement*).	J. S. Cunningham T. W. Torr	R. W. Wilson H. A. Brinker
DICKSON, DONALD R., Student, Purdue University, West Lafayette, Ind.	G. A. Hawkins (*ASME*) W. T. Miller	J. W. Raub (*ASME*) R. O. Sudy (*ASME*)

CANDIDATES

DIETER, GEORGE H., Sales Engr., The Fluor Corp., Ltd., Los Angeles, Calif.

FAULKNER, JOHN H., Secy-Treas., Langdon-Faulkner Co., Inc., Seattle, Wash.

FRANKLE, HARRY R., Pres.-Gen. Mgr., Air Control, Inc., Omaha, Nebr.

FREESTROM, ARTHUR, Sales Engr., O. Freiert Co., Alpena, Mich.

GESSELL, ELMER T., Sales Engr., W. E. Lewis & Co., Dallas, Tex.

GOINS, EDGAR H., Dist. Repr., Warren Webster & Co., Camden, N. J.

HOKANSON, CARL G., Owner-Engr., C. G. Hokanson Co., Los Angeles, Calif.

HOLDER, LEONARD H., Owner, April Showers Co., Washington, D. C.

HOWES, BRADFORD B., Sales Repr., Johnson Service Co., New York, N. Y.

JOHNSON, RUSSELL A., Student, University of Illinois, Urbana, Ill.

KAGEY, I. B., JR., Dist. Mgr., Carrier Corp., Atlanta, Ga. (*Reinstatement*)

KING, BRUCE A., JR., Engr., Giffels & Vallet, Detroit, Mich.

KNIGHT, JOHN T., JR., Consulting Engr., New Orleans, La.

KURTZ, OTTO, Mech. Engr., 3518 Grove St., Oakland, Calif.

LANGBERG, MARTIN, Vice-Pres., Carroll Sheet Metal Works, Inc., Winfield, L. I.

LANGDON, EDWIN H., Pres., Langdon-Faulkner Co., Inc., Seattle, Wash.

LOBSTEIN, MELVILLE C., Chief Operating Engr., Museum of Science & Industry, Chicago, Ill.

MARTIN, GEORGE D., Br. Mgr., Grinnell Co. of Pacific, San Francisco, Calif.

MELNICK, NICHOLAS A., Engr., G. M. Simonson, Consulting Engr., San Francisco, Calif.

MILLIGAN, DONALD G., Draftsman, G. G. Sharp, New York, N. Y.

MORAWECK, ALVIN H., JR., Secy.-Sales Engr., Abbott Engineering & Contracting Co., Newark, N. J.

RICHARDS, LESLIE V., Owner, Richards Oil Burner Sales & Service, Malden, Mass. (*Advancement*)

RITTLEMEYER, JOHN M., Owner, Rittlemeyer & Co., Atlanta, Ga.

ROBERTS, HARRY H., Air Cond. Engr., Arkansas Gas Corp., Shreveport, La.

RODEFFER, EDGAR W., Engr. & Supt., Burke & Son Co., Los Angeles, Calif.

ROWLEY, ROBERT K., Student, University of Minnesota, Minneapolis, Minn.

RUMBOLD, ALLAN H., Br. Mgr., Clow Gasteam Heating Co., Atlanta, Ga.

RUMSEY, JOHN L., Construction Engr., G. M. Simonson, Cons. Engr., San Francisco, Calif.

SCANDRETT, HAROLD R., Estimator, Pacific Gas & Electric Co., San Francisco, Calif.

SCHIMM, ELLIS L., Dept. Mgr., Macatee, Inc., Dallas, Texas.

SCOTT, WILLIAM P., JR., Vice-Pres., Scott Co., San Francisco, Calif. (*Advancement*)

STERNER, DOUGLAS S., Br. Mgr., Elec. Div., Barber-Colman Co., Rockford, Ill. (*Advancement*)

TERRY, SAMUEL W., Pres., Aladdin Heating Corp., Oakland, Calif.

TILFORD, LEO A., Owner & Mgr., Leo A. Tilford Co., Jackson, Mich.

WALKER, WYTHER F., Engr., Hayes Furnace & Mfg. Co., Los Angeles, Calif.

WELLS, EDWARD E., Engr., Aluminum Co. of Canada, Montreal, Que., Canada.

WEST, CHARLES H., JR., Vice-Pres., Massey, Wood, & West, Inc., Richmond, Va.

WHEELER, CHARLES A., Br. Mgr., Johnson Service Co., Milwaukee, Wis.

WISE, MASON W., Owner, M. W. Wise Co., Atlanta, Ga. (*Reinstatement*)

REFERENCES

Proposers	*Seconders*
M. H. Roths (*Non-member*)	A. B. Banowsky
M. L. Diver	C. A. McKinney
M. J. Hauan	C. W. May
H. T. Griffith	S. D. Peterson
P. R. Duitch	M. Olchoff
N. B. Delavan	R. A. Smith (*AIEE*)
F. C. Warren	B. F. McLouth
O. D. Marshall	T. D. Stafford
M. L. Brown	R. G. Lyford
R. K. Werner	G. A. Linskie
C. E. Bentley	L. H. Cochran
G. M. Simonson	N. H. Peterson
A. J. Hess	E. H. Kendall
H. English	H. B. Lauer
F. E. Spurney	F. C. Hackett
W. H. Littleford	F. A. Leser
H. J. Ryan	M. F. Rather
R. A. Wasson	F. E. W. Beebe
M. K. Fahnestock	E. L. Broderick
A. P. Kratz	R. B. Engdahl
C. L. Templin	S. W. Boyd
T. T. Tucker	L. F. Lawrence, Jr.
F. W. Johnson	F. C. Purcell
W. H. Old	F. X. Marzolf
W. P. Oster	O. E. Gamwill, Jr.
W. H. Dudley, Jr.	N. J. Helwick
G. M. Simonson	A. J. Bouey
C. E. Bentley	C. L. Peterson
G. A. Belsky	H. L. Baker, Jr.
J. A. Heller	B. Leventhal
M. J. Hauan	C. W. May
H. T. Griffith	S. D. Peterson
L. Seelig	C. M. Burnam, Jr.
J. J. Philippi	A. O. May
N. H. Peterson	G. M. Simonson
C. E. Bentley	L. H. Cochran
J. Gayner	C. E. Bentley
G. M. Simonson	C. L. Peterson
C. S. Pabst	V. J. Cucci
A. E. DeSomma	W. W. Timmis
H. L. Jones	R. C. Bonthron
W. F. Raymer, Jr.	F. D. McCann
H. M. Tarr	C. P. Yaglou
R. M. Nee	Philip Drinker
C. L. Templin	L. F. Lawrence, Jr.
A. H. Koch	R. F. Hahn
L. C. Mabley	H. T. Goss (*Non-member*)
R. C. Bailey	A. L. Zwally
L. Hungerford	W. O. Stewart
H. H. Douglas	A. J. Hess
R. C. Jordan	C. E. Lund
A. B. Algren	F. B. Rowley
C. L. Templin	A. H. Koch
L. L. Barnes	L. F. Lawrence, Jr.
G. M. Simonson	C. E. Bentley
J. Gayner	N. H. Peterson
C. E. Bentley	L. H. Cochran
N. H. Peterson	G. M. Simonson
J. A. Bishop	O. H. Mehl
M. W. Brown	L. L. Landauer
C. E. Bentley	G. M. Simonson
J. F. Kooistra	A. J. Bouey
T. T. Tucker	F. E. Baird
C. L. Templin	S. W. Boyd
C. E. Bentley	James Gayner
G. M. Simonson	N. H. Peterson
F. C. Warren	S. J. Dempsey
Harry Christenson	R. A. Parsons
A. J. Hess	H. H. Douglas
W. O. Stewart	H. H. Bullock
G. L. Wiggs	A. F. Lamontagne
W. W. Timmins (*Non-member*)	W. E. Baxter
E. C. Webb	J. A. Johnston
T. H. Smoot	W. E. Carle
B. E. Landes	H. E. Drain
C. Clegg	R. H. Schnell
C. L. Templin	A. H. Koch
L. L. Barnes	L. F. Lawrence, Jr.

Candidates Elected

In the past issues of the JOURNAL of the Society the names of the following men were listed as Candidates for Membership. The membership grade of each Candidate has been assigned by the Committee on Admission and Advancement and balloted upon by the Council. We are now instructed by the Council to post herewith, as required by Art. B-III, Sec. 8, of the By-Laws, the following list of candidates elected:

MEMBERS

ANDERSON, EDWIN L., Asst. Industrial Hygienist, State Health Dept., Hartford, Conn. (*Reinstatement*)

CRUMLEY, MEARL T., Engineer, Robert & Co., Atlanta, Ga.

DISTEL, ROBERT E., Mgr., Distel Htg. Equipment Co., Lansing, Mich. (*Advancement*)

EHRENZELLER, ADOLPH, Htg. Engr. & Contractor, A. Ehrenzeller, Dorchester, Mass. (*Reinstatement*)

FANNING, ERROLL C., Engr., Atlas Heating & Ventilating Co., Ltd., San Francisco, Calif.

FRIEND, WALTER F., Mech. Engr., Ebasco Services, Inc., New York, N. Y.

FRITZBERG, LAWRENCE H., Asst. to Chief Engr., Aerofin Corp., Syracuse, N. Y. (*Reinstatement*)

GARRARD, WALTER M., Mfrs. Agent, Garrard & Co., Atlanta, Ga.

GENONE, HENRY W., Vice-Pres. charge of Engrg., Dinkler Hotels Co., Inc., Atlanta, Ga.

GOODMAN, WILLIAM, Consulting Engr., The Trane Co., La-Crosse, Wis.

GORBANDT, EVERETT T., Partner, Crawley-Gorbandt Co., Atlanta, Ga.

HAMIG, LOUIS L., Engr., John D. Falvey-Consulting Engr., St. Louis, Mo. (*Advancement*)

HOOD, LESLIE A., Asst. Sales Mgr., Trane Co. of Canada, Ltd., Montreal, Que., Canada (*Reinstatement*)

JORDAN, FRANCIS O., Specification Engr., Albert Kahn, Inc., Detroit, Mich.

LAUER, RODNEY F., Dist. Sales Mgr., York Ice Machinery Corp., Philadelphia, Pa. (*Advancement*)

LESER, FREDERICK A., Dist. Mgr., Ilg Electric Ventilating Co., Washington, D. C. (*Advancement*)

ROEDER, WINFIELD, Br. Mgr., American Blower Corp., New Haven, Conn.

SHAPIRO, MORRIS, Mech. Engr., U. S. Housing Authority, Washington, D. C.

SMITH, HAROLD C., Mgr., Equipment Dept., The Darby Corp., Kansas City, Kans.

STECKHAN, LOUIS, Mgr., Engrg. Dept., Crane Co., St. Louis, Mo. (*Reinstatement*)

WEEKES, ROY W., Gas Htg. Engr., Iowa City Light & Power Co., Iowa City, Ia.

WHITE, WILLIAM E., Engrg. & Sales, The Darby Corp., Kansas City, Kans.

ZIEL, HERBERT E., Chief Engr., Albert Kahn Inc., Detroit, Mich. (*Reinstatement*)

ASSOCIATES

BURRITT, EDWARD D., JR., Sales Engr., Burnham Boiler Corp., Cambridge, Mass.

CLARKE, JOHN H., Marine Engr., Maritime Commission, Washington, D. C.

FARLEY, WILLOUGHBY S., Contractor & Engr., Danville, Va. (*Reinstatement*)

FERRIS, ARTHUR L., Sales Engr., Hart & Cooley Mfg. Co., Fort Erie, N. Ontario, Canada.

GEZARI, ZVI, Mechanic, Adams Engineering Co., New York, N. Y.

HUELSMANN, A. G., Supt., The Powers Regulator Co., Cincinnati, Ohio.

KEMP, GORDON C., Mgr., Htg. Dept., Chatham Malleable & Steel Products, Ltd., Toronto, Ont., Canada.

LERICHE, RAYMOND E., Br. Mgr., Central Equipment Co., Portland, Oregon.

LYON, DOUGLAS MCCLURE, Partner, Douglas McClure Lyon, Syracuse, N. Y.

MCGINN, GEORGE F., Owner, Industrial Engineering Co., Baltimore, Md.

O'SHEA, JOHN J., Branch Manager, Buffalo Forge Co., Atlanta, Ga.

REILLY, PHILIP H., JR., Office Mgr., Heating Equipment Co., San Francisco, Calif.

ROBIN, RICHARD C., Owner, Richard C. Robin, Sales & Engrg., Boston, Mass.

SPROTT, JOHN I., Sales, Ray Oil Burner Co., San Francisco, Calif.

STERN, EDWARD J., Sales Engr., Armstrong Cork Co., Dallas, Tex.

TOBIN, THEODORE A., Dist. Sales Mgr., The Marley Co., Atlanta, Ga.

WAHLIN, BERNARD J., Field Repr., General Electric Co., Bloomfield, N. J.

ZINTEL, GEORGE V., Sales Engr., Himelblau, Byfield & Co., Chicago, Ill.

JUNIORS

BARNEY, WILLIAM J., Sales Repr., Hoffman Specialty Co., Inc., Waterbury, Conn.

BOYD, ROBERT L., JR., Sales Engr., Houston Lighting & Power Co., Houston, Tex.

BREWER, FRANK M., Mech. Draftsman, Robert & Co., Inc., Atlanta, Ga.

POST, NICHOLAS, Army Ordnance Inspector, War Dept., St. Louis Ordnance Dist., St. Louis, Mo.

RAMONEDA, ENRIQUE, Contractor & Repr., Clarage Fan Co., Mexico, D. F., Mexico.

SMITH, DAVID J., Sales Engr., Walter H. Eagan Co., Philadelphia, Pa.

STRANSKY, MALCOLM W., Engr., Heating Equipment Co., San Francisco, Calif.

SWAIN, DOUGLAS S., Sales Engr., Trane Co. of Canada, Ltd., Winnipeg, Man., Canada.

WARNER, CECIL F., Instructor, Mech. Engrg., Lehigh University, Bethlehem, Pa.

STUDENTS

HAYES, ORRIS J., Student, University of Minnesota, Minneapolis, Minn.

MATTHIES, LEO A., Student, University of Minnesota, Minneapolis, Minn.

TUPPER, EDWARD B., Student, University of Minnesota, Minneapolis, Minn.